THE

Modern Gas-Engine

AND THE

Gas-Producer

BY

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PREFACE

The importance of effecting the greatest possible improvement in the economy with which the national fuel-resources are utilized will be apparent when it is considered that under existing conditions, and with the demands increasing at the rate they have since active coal-mining first became an established industry, the present estimated coal-reserve, about 2,000 billion tons, will have been absorbed before the end of the next century. The yearly consumption is now 500,000,000 tons and it is increasing at the rate of doubling itself every ten years.

Of the total amount mined, so far, due to exacting requirements with regard to the quality of the marketable fuel, and due to wasteful methods of mining it, there has been left in the ground, inaccessible for future use, or wasted as unmarketable product, an amount even greater than that actually rendered useful.

The portion of the fuel consumed for industrial purposes, for generating power, or for metallurgical use, which is by far, the greater portion of the fuel actually rendered useful, has, according to modern standards, been utilized particularly inefficiently; but a tendency toward a strict reform in this respect is afoot, and the problem at present is how to utilize, not only the energy in the good grades of fuel to the greatest possible extent, but also to find new ways for an effective utilization of inferior grades hitherto wasted.

The gas-engine is looked upon as a means for effecting reform in the one respect as well as in the other, and, although the final development of the engine has probably not as yet been seen, its theory indicates at present how far an increased economy may be hoped for, just as the steam-engine theory, in the days of Watt, indicated the final limit for efficiency of the steam-engine cycle, toward which the actual performance of the engine has gradually, by steps, approached closer.

In the earlier stages of its development, the gas-engine was employed principally because of the convenience with which it could be installed for generating, economically, from high-priced fuels, limited demands for power; and its increasing employment on similar grounds is still insured. Any economy derived under these conditions is due to the decided economy of the engine, or due to the low cost of its installation, not to any economy with regard to the cost of the fuels consumed. Of late years, however, a rapid and fortunate transformation with respect to the available gas-engine fuels has been effected, in that the retorted coal gas and other less available fuels have been replaced by inexpensive fuels, such as producer-gas, carbureted water gas, crude oil, etc.; and blast-furnace gas and coke-oven gas have become available industrial fuels. Under these conditions, and particularly since the development of the engine into its modern types, the economy of the gas-engine has been proven, in practice as in theory, to exceed, by far, any the steam-engine could well be expected to give under the present temperature range of its cycle.

As a consequence of its rapid development and growing importance, some speculation has been indulged in as to the possibility that the gas-engine ever will displace the old prime mover, in the face of the inconsistencies and difficulties the former often exhibits. As to this question, it may be reassuring to assume that history will repeat itself, in that, when new forces for deriving a cheapened supply of work have been discovered, new fields for demand have been opened, which, finally, have put an even greater demand on the old as well as the new force for an increased output. The innate power for economy of the gas-engine is, however, such that in spite of difficulties, imagined or real, its future usefulness cannot well be doubted. Its position may, in certain uses, be disputed, but such conditions prevail in many fields—and these will no doubt become extended in the future—that due regard for a proper economy simply demands its service, and leaves it in these fields essentially without a competitor.

That the interest in the gas-engine, due to its availability and economy, is ample and general, not only among engine-builders and users in this country, but, perhaps, in an even higher degree

on the Continent, is shown by the steady increase in the number of engines installed; and it is toward this fact the author is looking for justification for the appearance of the present volume.

The main matter the book contains consisted originally of a collection of notes arranged with the idea of presenting, for personal use, as clearly and as directly as possible, any of the numerous questions pertaining to the gas-engine that, in practice, often come up and require immediate answers. To present this collection properly in a book designed to be of use to others in a situation similar to that of the author, or, in fact, to any one interested in the subject, the notes have been extended and supplemented by necessary elementary matter, to make them readable for others.

Throughout the preparation of the volume, the object has been to economize as much as possible with the student's patience, in reading it, and with the user's time, in looking up any special subject that practice may bring forth; still preserving the ample general subject in a tolerably complete form.

An earnest student, recently propounded the rather perplexing question: "What book can I get that will tell me everything about gas-engines, but *with few square roots?*" the italicized expression having reference to mathematical work in general. Of course, few books ever pretended to tell everything about any subject, and, certainly, no book could tell even a small part of everything about the gas-engine, intelligently and completely, without the use of some *square roots*. This proposition—it being next to an axiom, made clear, it was readily agreed that some serviceable formulas would actually fill a long-felt want.

The formulas are often what the designer requires most, and, of course, he is anxious to know exactly the conditions under which they have been obtained, in order to use them with confidence. But it is not the designer only who will have use for the formulas. The engineer who wants to know something about his engine needs them as much; and if he fails to learn how to apply some simple formula to the facts his engines disclose, he will miss, in a great measure, the power of controlling, to the best advantage, the events that influence the operation of his plant.

All of the formulas of the first chapter are the old and standard

ones, collected from different sources, though the method employed for their derivation is in some respects new and original. The formulas directly relating to the gas-engine are not, all, old—some are new, and some have been supplied with coefficients which make them more suitable for practical use, but they all agree with the old and standard ones. Nearly all the formulas have been taken directly from the author's record-books, and, judging from the present appearance of these, the indications are that the formulas have, on more than one occasion, been used and tried in practice; and it is believed they all are reliable.

In determining sizes and dimensions of parts of a design, which by a designer is, of course, always, in the first hand, done independently of what others have done, it is often of satisfaction to find that our final dimensions agree with those commonly used, which may have become more or less standard. In order to supply the means whereby any young designer's confidence in his work may be enhanced, tables of gas-engine details actually used in practice have, in several instances, been submitted.

In order to acquire a working knowledge of the subject, the general types of gas-engines in successful operation at present should be studied, in connection with the general laws on which their design and operation must be based, because from these types, according to the laws of evolution, there naturally will be developed the successful types of the future. Chapter XIV is devoted to this branch of the subject, and the author here takes occasion to express his appreciation of the generosity with which engine-builders have placed illustrations and information at his disposal.

With the subject of the gas-engine there is at present allied that of the gas-producer, so closely, that a full familiarity with the former would, in the main, seem purposeless without a working knowledge of the latter. The individual gas-producer types in general use are many; perhaps too many, considering that the difference between different types is often slight, often immaterial, and justified only because the producer is as yet in a state of evolution. The main types are not so many, and, in treating of the subject, the purpose has been to limit the discussion to those

producer-types which possess the most interesting and fundamental features. To do the important subject full justice, however, there would be required far more space than has been assigned to it in the last chapter of this volume.

In completing the work, and in order to bring as many of the general facts about the gas-engine as possible inside its narrow confines, all available sources for information have been freely consulted, and a general list of the more important of these is appended among the final pages of the book.

A. M. LEVIN.

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CHAPTER I

INTRODUCTION TO THERMODYNAMICS

Fundamental Equations.—The only fundamental variables with which Thermodynamics is concerned are pressures, volumes, and heat; and all substances possess these properties. The temperature, which also characterizes the thermal condition of every substance, is determined by its pressure and volume, exclusively, and may therefore be considered a secondary variable. Some objects have acquired motion, or have stored mechanical energy, which, according to the first principle of Thermodynamics, is only another manifestation of heat, and, therefore, motion is also a secondary variable.

The fundamental relations between the variables of Thermodynamics are two; of which the first may be expressed as follows:

The internal heat of every substance depends, exclusively, on its pressure and volume; and the second is found in the following principle:

When heat is transmitted to a substance whose volume increases on account of a rise in temperature, part of it is absorbed as internal heat, and part is transformed into external work which is utilized in effecting an increased space for the substance, against the external pressure it may bear on its surface. This law may be expressed by the equation

$$Q = U + \frac{1}{J}\int_{V_1}^{V_2} P\,d\,V, \quad . \quad . \quad . \quad . \quad (1)$$

Q being the heat transmitted,

U the gain in internal heat,

$\int_{V_1}^{V_2} P\,d\,V$ the external work performed by the substance,

and J the mechanical equivalent of heat. See page 9.

The expression $\int_{V_1}^{V_2} P\,d\,V$ (the integral of P times $d\,V$, between the limits V_1 and V_2) we retain from the calculus, simply, in this connection, as a conventional symbol for the work performed by an expanding substance, when its volume increases from V_1 to V_2.

As it will be necessary to refer to identically the same symbol at several occasions in the following, it may be well to make the meaning of it clear at this place.

Fig. 1 is a graphical representation of the changes in volume and pressure that take place during the expansion of a working substance; for instance, of a quantity of air. V_1 and P_1 represent the initial volume and pressure of the substance and V_2 its final volume. The pressure is assumed to change according to the line $B\,C$, while the volume gradually changes from A to D so that when the volume is, for instance, V, the pressure is P, and when the volume becomes V_2 the pressure becomes P_2. While the volume increases a small amount, represented by $\delta\,V$, the pressure may, without a great error, be considered to be constant and equal to P, wherefore the work done during this elementary expansion is $P \times \delta V$, and it may be similarly expressed for each small increment in the volume, between V_1 and V_2. Hence, by summing together all the small rectangles which may be inscribed between the base-line $A\,D$ and the line $B\,C$, with V_1 and V_2 as limits, we get, approximately, the total work done during the expansion from V_1 to V_2.

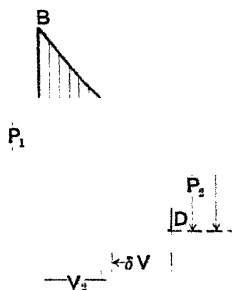



FIG.

The error involved by representing the work due to each elementary expansion by a rectangle disappears when $\delta\,V$ is made infinitely small, so that the sum of all the infinitely narrow rectangles, between the limit V_1 and V_2, becomes, then, the true work done during the entire expansion. This work is represented by the area $A\,B\,C\,D = F$.

The integral, $\int_{V_1}^{V_2} P\,dV$, is an expression for the sum of all the infinitely narrow rectangles $P \times \partial V$,* that can be inscribed between the limits V_1 and V_2, and represents, therefore, the area of the figure $A\,B\,C\,D$, or the work of expansion of the substance, when the pressure changes according to the arbitrary line $B\,C$, while the volume changes from A to D.

The pressure may, during an expansion, change in such a way that the product $P \times V$ remains a constant quantity $= K$, and in that case we write

$$P\,V = K, \text{ or } P = \frac{K}{V}.$$

Inserting this value for P in the integral we get the expression

$$F = K \int_{V_1}^{V_2} \frac{d\,V}{V}.$$

An integral of this appearance—it may refer to pressures, temperatures, or any variable quantity as well as volumes—is of fundamental form and can be integrated into a definite result. Written in a general way, by changing V for x, the equation becomes by integration

$$F = K \int_{x_1}^{x_2} \frac{d\,x}{x} = K\,(log_e\,x_2 - log_e\,x_1) = K\,log_e \frac{x_2}{x_1}; \qquad (2)$$

log_e denoting the hyperbolic logarithm.

The **Second Principle of Thermodynamics.**—Carnot stated the Second Principle of Thermodynamics as follows:

Assuming that a working substance be, successively, put in communication with a source of heat and with a source for refrigeration (a source for abstracting heat) in such a manner that no heat is received or abstracted, except from or by either of these sources, and so that, while receiving or rejecting heat, its temperature remains constant and equal to that of the source with which it, for the time being, is in communication; if the substance finally

* Two different notations, dV and ∂V, for the width of the elementary rectangles have been employed. This is according to common usage. Behind the sign of integration, PdV is an infinitely narrow rectangle, while, on the drawing, the notation $P\partial V$ represents an elementary rectangle of finite width.

returns to its initial state, then the ratio of the heat which it has abstracted from the source of heat to that which it has rejected to the refrigerator is independent of the nature of the substance, and depends only on the temperature-limits between which it is working.

As every heat-engine, in operation, receives heat from a source of heat at a higher temperature, and rejects part of this heat at a lower temperature, after having transformed a part of it into work, this law may also be expressed thus: The efficiency of an engine working in a closed reversible cycle is independent of the nature of the working substance, and depends only on the temperature limits between which it is working.

Fig. 2 represents the changes in a substance when it is working in conformity with Carnot's cycle. While in communication with the source of heat H, during which time its temperature is constant and equal to T_1, the pressure and volume of the substance change according to the line $B_1\,C_1$, pressures being measured with reference to the axis $O\,V$ and volumes with reference to the axis OP. The line $C_1\,D_1$ represents its adiabatic* expansion from the temperature T_1 to the temperature T_2, the line $D_1\,A_1$ its isothermal† compression, at the temperature T_2, when in communication with the refrigerator R, and $A_1\,B_1$ its adiabatic compression from the temperature T_2 to the temperature T_1.

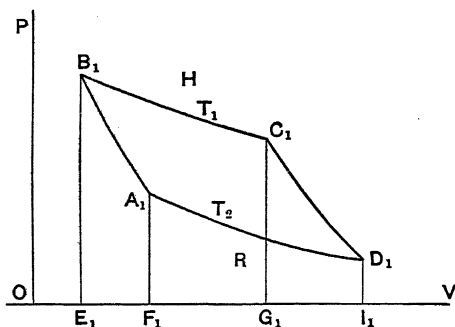



FIG. 2.

When expanding from B_1 to C_1 the substance receives heat while generating the work represented by the area $B_1\,C_1\,G_1\,E_1$,

* An expansion or compression is called adiabatic when, during the process, no heat is absorbed or rejected by the substance.

† An expansion or compression is called isothermal when the temperature of the substance remains constant during the process.

and while expanding from C_1 to D_1 it generates the work $C_1\,D_1\,I_1\,G_1$. On the other hand, when passing from D_1 to A_1 it rejects the energy $D_1\,I_1\,F_1\,A_1$, and when passing from A_1 to B_1 it absorbs the energy $A_1\,F_1\,E_1\,B_1$. The net work realized during the cycle is, therefore, represented by the area $A_1\,B_1\,C_1\,D_1$.

The absolute temperatures of the substance, while passing from B_1 to C_1 and from D_1 to A_1, Fig. 2, are represented in Fig. 3 by the heights of the lines $I\,C$ and $G\,D$ above the absolute zero-line $O\,\phi$. The lines $B\,E$ and $C\,F$ are drawn perpendicular to the lines $I\,C$ and $G\,D$, enclosing an area, $A\,B\,C\,D$, which is made to represent, to a certain scale, the work generated during the cycle. Each side of the rectangle $A\,B\,C\,D$ represents, then, the same event as the corresponding side of the figure $A_1\,B_1\,C_1\,D_1$. At H the substance receives the heat Q_1, at the temperature T_1, and at R it rejects the heat Q_2, at the temperature T_2; $Q_1 - Q_2$ being the heat transformed into work.

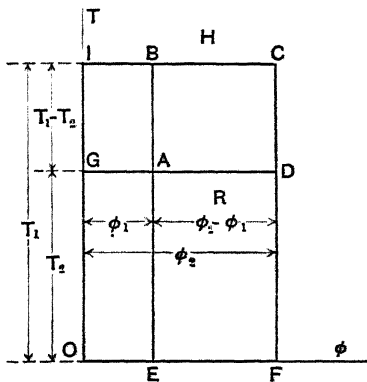



FIG. 3.

If we call the width of the rectangle, according to the notation in the figure, $\phi_2 - \phi_1$ we have

$$(\phi_2 - \phi_1)\,(T_1 - T_2) = Q_1 - Q_2 = \frac{1}{J}L,$$

$$\text{or } J\,(\phi_2 - \phi_1) = \frac{L}{T_1 - T_2}. \quad . \quad . \quad . \quad . \quad (3)$$

L being the work performed, expressed in foot-pounds.

The quantity $\phi_2 - \phi_1$ is the change of the so-called entropy of the working substance; and the similarity between the part played by this property of heat at the transformation of heat-energy into work, through declining temperatures, and the part played by a lifted weight when its potential energy is transformed into work, through declining elevations, is suggested by the

similarity between the equations expressing the rate at which work is being done in each case.

If a weight of W pounds drops from a height H_1 to a height H_2 we have, namely,

$$W = \frac{L}{H_1 - H_2};$$

an equation which is identical with 3.

The working substance has absorbed, at C, Fig. 3, the entropy ϕ_2, and during its adiabatic expansion from C to D, while its temperature is being lowered from T_1 to T_2, this property remains unchanged. While being compressed at a constant temperature T_2, between the points D to A, it rejects the entropy $\phi_2 - \phi_1$, and it returns to B, through adiabatic compression, with the entropy ϕ_1. Thus it abstracts, during the complete cycle, from the source of heat the entropy $\phi_2 - \phi_1$. This interval of entropy would do the most effective work by moving from the temperature-level $T = T_1$ to the level $T = 0$, and the heat transferred into work would then be the greatest conceivable, viz.:

$$Q_1 = (\phi_2 - \phi_1)\ T_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

On account of lack of facilities to reject the entropy at the zero-level, we cannot, however, carry the cycle so far, but must, of necessity, confine it above a higher level where heat can be rejected; for instance, at the level T_2.

Defining the efficiency of an engine as the ratio of the heat transformed into work to the heat supplied, and denoting the efficiency by E, we have:

$$E = \frac{L}{J\,Q_1} = \frac{Q_1 - Q_2}{Q_1} = \frac{(\phi_2 - \phi_1)\ (T_1 - T_2)}{(\phi_2 - \phi_1)\ T_1}$$

$$\text{or } E = \frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\text{From this we get } \frac{Q_1}{Q_2} = \frac{(\phi_2 - \phi_1)\ T_1}{(\phi_2 - \phi_1)\ T_2}, \quad . \quad . \quad . \quad (6)$$

$$\text{and } Q_2 = (\phi_2 - \phi_1)\ T_2 \quad . \quad . \quad . \quad (7)$$

Hence, $Q_1 - Q_2$ (the heat which has been converted into work) being represented by the rectangle $A\,B\,C\,D$, Fig. 3, the heat

supplied, Q_1, is represented by the rectangle $EBCF$ and the heat rejected by the rectangle $EA\ DF$.

From equations 4 and 7 it is evident that the heating-value absorbed by a working substance is proportional to the product of its temperature and increase of entropy, and that, when heat is transformed into work by adiabatic expansion, the entropy of the substance remains constant and its absolute temperature decreases in the same proportion as its potential energy.

Expressions for Entropy.—When the temperature remains unchanged while the substance receives heat, then the increase in entropy may be defined as: the total heat received divided by the absolute temperature at which it was received.

The heat of vaporization in steam, or any vapor, being received at a constant temperature we have, therefore,

$$\phi_2 - \phi_1 = \frac{Q}{T} = \frac{rx}{T};$$

r being the heat of vaporization, and x the percentage of dry steam or vapor.

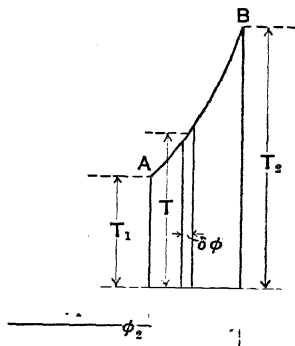
If, however, the temperature of the substance increases while heat is absorbed, as is the case when water is heated from the temperature of the feed to the temperature at which it is vaporized, or when heat is transmitted to a body of gas or air, then the expression for the change in its entropy becomes more complex.

If by δQ be denoted the heat which is absorbed by the substance while its entropy increases an elementary amount $\delta\phi$, Fig. 4, then we have

$$\delta\phi = \frac{\delta Q}{T};$$

T being the average absolute temperature during the increment.

FIG. 4.

Let the curve AB, Fig. 4, represent the change in entropy and temperature between the temperature-limits T_1 and T_2, then δQ is represented by the area of the elementary rectangles, such as

$T \times \delta \phi$. Hence, if we sum together the ratios $\frac{\delta Q}{T}$ (ratio of area to height) of all the elementary rectangles, within the limits T_2 and T_1, we get the total increment of entropy. Thus,

$$\phi_2 - \phi_1 = \int_{T_1}^{T_2} \frac{dQ}{T}.$$

The unit weight of a substance absorbs for each degree rise in temperature a definite amount of heat, called its specific heat. This heat is not always the same at very high temperatures as it is at low ones, but for ordinary temperatures it is approximately constant.

Let the specific heat be called c, and let δT signify the small increase in temperature that corresponds to the increment δQ in heat absorbed, then

$$\delta Q = c\, \delta T,$$

$$\text{and hence } \phi_2 - \phi_1 = \int_{T_1}^{T_2} c \frac{dT}{T},$$

or, if c be assumed to be a constant quantity, we get

$$\phi_2 - \phi_1 = c \int_{T_1}^{T_2} \frac{dT}{T}.$$

By integration of this expression, similarly to that of the general equation 2, we get

$$\phi_2 - \phi_1 = c \log_e T_2 - c \log_e T_1,$$

$$\text{or } \phi_2 - \phi_1 = c \log_e \frac{T_2}{T_1}. \quad \quad (8)$$

The entropy of water is, in steam tables, generally given above 32° F., and is denoted by θ.

We have, thus,

$$\theta = c \log_e \frac{460 + t}{492}, \text{ approximately;}$$

t being a temperature above zero on the Fahrenheit scale. A mean value for c would be 1.013.

Notations and Definitions.—The formulas derived in the following refer, strictly, only to perfect gases. Practically, they are, however, applicable to air or fuel-gas, and even to steam

when superheated to a degree such that, after the expansion has taken place, the final steam is approximately in a dry condition. Air being a practically perfect gas, we shall refer to the working agent as air, in preference to using the word "gas," which appears less well defined.

The notations used are principally the following:

ϵ = the coefficient of expansion of air, or the increase in its volume for one degree's rise in temperature when the pressure is constant $= \frac{1}{491.2} = 0.002036$.

$a = \frac{1}{\epsilon} - 32^\circ$ = the distance to the absolute zero, below zero of the Fahrenheit scale, in degrees = 459.2.

c_p = the specific heat of air at a constant pressure = 0.2375.

c_v = the specific heat of air at a constant volume = 0.1689.

$n = \frac{c_p}{c_v} = 1.406$.

D_o = the weight of one cubic foot of air of a temperature of 32° F., and at a pressure of one atmosphere $= \frac{1}{12.39}$ 0.080728 pounds.

P_o = the pressure of the atmosphere, per square foot, = 2116.3 pounds at a barometric pressure of 29.92 inches ($760^m/_m$) mercury.

$R = \frac{\epsilon P_o}{D_o}$.

V = the volume of one pound of the working substance, in cubic feet.

P = the pressure of the working substance, in pounds per square foot.

$T = a + t$ = the absolute temperature of the working substance.

t = the temperature on the Fahrenheit scale.

Principal Laws of Thermodynamics.—Heat and mechanical energy are mutually convertible in a certain fixed ratio, which, according to the latest investigations, is such that one heat-unit is equal to 778 foot-pounds. The mechanical equivalent of heat is, for convenience, according to general practice, expressed by

the symbol J. The heat equivalent of mechanical work is, therefore, $\frac{1}{J}$.

The law of Mariotte is expressed as follows: For any change in the volume of a given mass of a perfect gas, at a constant temperature, the product of pressure and volume is constant, that is: $V \times P$ = constant.

The law of Gay-Lussac is expressed as follows: The volume of a perfect gas increases, when the pressure is constant, proportionately with the increase in its absolute temperature.

$$\text{Hence, } V_1 = V_o(1 + \epsilon[t - 32°]), \quad . \quad . \quad (9)$$

$$\text{or } V_1 = V_o \epsilon (a + t) = V_o \epsilon T; \quad . \quad . \quad (10)$$

$t - 32$ being the range of temperature above that of melting ice and V_o the volume of the gas at the same temperature.

For the determination of the volume, pressure or temperature of a perfect gas, under any condition, these laws may be combined as follows:

Assume that one pound of air has at three occasions the following volumes, pressures, and temperatures.

	Volume.	Pressure.	Temperature.
(1)	V	P	t
(2)	V_o	P	32
(3)	$\frac{1}{D_o}$	P_o	32°

Thus: (1) and (2) combined give $\frac{V}{V_o} = \epsilon(a + t)$

(2) and (3) combined give $V_o D_o = \frac{P_o}{P}$

and these equations combined give $V P = \frac{\epsilon P_o}{D_o}(a + t) \quad . \quad (11)$

$$\text{or } VP = R(a + t) = RT. \quad . \quad . \quad . \quad . \quad (12)$$

By transformation there is obtained

$$\frac{T}{V} = \frac{P}{R} \text{ and } \frac{T}{P} = \frac{V}{R}. \quad . \quad . \quad . \quad . \quad (13)$$

The coefficient R may be written

$$R = \frac{1}{D_o} P_o \epsilon.$$

$\frac{1}{D_o}$ being the volume of one pound of free air, and ϵ the percentage this volume expands for each degree increase in temperature. R may therefore be defined as the amount of external work that must be performed, when the temperature of one pound of air is increased one degree Fahrenheit under a constant atmospheric pressure. Expressed in figures this work is:

$$R = \frac{12.39 \times 2116.3}{491.2} = 53.37 \text{ foot-pounds.}$$

Specific Heat at Constant Volume and at Constant Pressure.— The simple relation existing between the specific heat of a perfect gas at constant volume and its specific heat at a constant pressure may be derived as follows:

Assume that heat is transmitted to two identical quantities of air, each of one pound, so as to increase the temperature of each from T_i to T_f. In the first case heat is transmitted at a constant pressure, and in the second at a constant volume.

Since, in the first case, the volume of the air changes, say, from V_i to V_f, therefore, the total heat absorbed must partly have been used for doing the work

$$(V_f - V_i)\, P,$$

and partly absorbed as internal heat, U.

The heat-balance will, therefore, appear as follows:

$$c_p\,(T_f - T_i) = U + \frac{1}{J}\,(V_f - V_i)\, P.$$

On the other hand, when the air is heated at a constant volume there is no work performed, and the heat-balance must be written

$$c_v\,(T_f - T_i) = U.$$

The internal heat, U, being in both cases the same, the equations may be combined into

$$c_p\,(T_f - T_i) = c_v\,(T_f - T_i) + \frac{1}{J}\,(V_f - V_i)\, P.$$

This equation may be written

$$c_p\,(T_f - T_i) = c_v\,(T_f - T_i) + \frac{1}{J} R\,(T_f - T_i)$$

Hence, $$c_p - c_v = \frac{R}{J}. \qquad (14)$$

The specific heat c_v corresponds to the heating-value that the unit weight of air, when being heated, actually stores as heat, per degree increase in temperature, and the specific heat c_p is equal to the former, plus a certain heating-value necessary for doing work in expanding the air.

When air is compressed adiabatically, then all the work expended for its compression is converted into internal heat, and when it expands adiabatically all the work performed by it is obtained at the expense of internal heat. If the temperature, in either case, were changed from T_i to T_f then the gain in internal heat would be

$$Q = U = c_v (T_f - T_i), \quad . \quad . \quad . \quad . \quad (15)$$

and the corresponding work

$$L = J Q = J c_v (T_f - T_i). \quad . \quad . \quad . \quad . \quad (16)$$

Accordingly, when the initial temperature of the air is higher than the final then the air gains *negative heat-energy*, or, in other words, it performs work.

General Equations for the Transformation of Heat-Energy.— The amount of heat that a gas absorbs at a given change in its temperature depends on the two quantities: its specific heat at constant pressure and its specific heat at constant volume. The total entropy which the gas contains may be considered, therefore, to be due to two separate transformations, one at a constant pressure and the other at a constant volume. The curve $A B$, Fig. 5, represents the change in the state of heat of one pound of air when its temperature changes from T_1 to T_x while the pressure remains constant, and the curve $B C$ represents the change in the state of its heat, when its

FIG. 5.

temperature changes from T_x to T_2 while the volume remains constant. The heat supplied during the first change, at constant pressure, is represented by the area $a_o\, A\, B\, b_o$ and the heat supplied during the second change, at constant volume, is represented by the area $c_o\, C\, B\, b_o$. Assuming that the specific heat at constant pressure, c_p, is a constant quantity, then we have seen that the expression for the change in entropy during the transformation from A to B is expressed by equation 8

$$\phi_x - \phi_1 = c_p \log_e \frac{T_x}{T_1} \quad . \quad . \quad . \quad . \quad . \quad (17)$$

Similarly, the change in entropy due to the transformation from B to C, assuming the specific heat at constant volume to be constant, is

$$\phi_x - \phi_2 = c_v \log_e \frac{T_x}{T_2}. \quad . \quad . \quad . \quad . \quad . \quad (18)$$

For a perfect gas we have (equation 12)

$$P\,V = R\,T,$$

$$\text{giving } \frac{P_x V_x}{P_1 V_1} = \frac{T_x}{T_1} \text{ and } \frac{P_x V_x}{P_2 V_2} = \frac{T_x}{T_2}.$$

Assuming, successively, the pressure and the volume to be constant quantities during the transformation represented by these equations, thus:

$$\text{at constant pressure } \frac{V_x}{V_1} = \frac{T_x}{T_1},$$

$$\text{and at constant volume } \frac{P_x}{P_2} = \frac{T_x}{T_2}.$$

But, according to the notation in Fig. 5, we have

$$V_x = V_2 \text{ and } P_x = P_1.$$

$$\text{Therefore, } \frac{V_2}{V_1} = \frac{T_x}{T_1}, \quad . \quad . \quad . \quad . \quad . \quad (19)$$

$$\text{and } \frac{P_1}{P_2} = \frac{T_x}{T_2}. \quad . \quad . \quad . \quad . \quad . \quad (20)$$

$$\text{Hence, } \log_e T_x = \log_e \frac{V_2}{V_1} + \log_e T_1, \quad . \quad . \quad . \quad (21)$$

$$\text{and } \log_e T_x = \log_e \frac{P_1}{P_2} + \log_e T_2. \quad . \quad . \quad . \quad (22)$$

The increase in entropy for a change from A to C we find in the difference between equations 17 and 18, thus:

$$\phi_2 - \phi_1 = c_p \log_e \frac{T_x}{T_1} - c_v \log_e \frac{T_x}{T_2}$$

Substituting in this equation, successively, (1st), the values of $\frac{T_x}{T_1}$ and $\frac{T_x}{T_2}$ from 19 and 20; (2nd), the value of $\log_e T_x$ from 21; and (3rd), the value of $\log_e T_x$ from 22, we get

$$\phi_2 - \phi_1 = c_p \log_e \frac{V_2}{V_1} - c_v \log_e \frac{P_1}{P_2} \quad . \quad . \quad . \quad . \quad . \quad (23)$$

$$= c_v \log_e \frac{T_2}{T_1} + (c_p - c_v) \log_e \frac{V_2}{V_1} \quad . \quad . \quad . \quad (24)$$

$$= c_p \log_e \frac{T_2}{T_1} + (c_p - c_v) \log_e \frac{P_1}{P_2} \quad . \quad . \quad . \quad (25)$$

For the following special cases relating to changes in the condition of a given quantity of air, we assume the weight of the air to be one pound.

V_i, P_i and t_i denote in each case the initial volume, pressure, and temperature of the air.

V_f, P_f and t_f its final volume, pressure, and temperature,

Q is the amount of heat absorbed or rejected,

and L the work consumed or generated.

Changes in the Temperature of One Pound of Air, at Constant Volume.—The heat required to increase the temperature of one pound of air at a constant volume is

$$Q = c_v (t_f - t_i).$$

When the temperature decreases, then the right-hand side of the equation becomes negative, and hence the same quantity of heat is in that case rejected.

According to equation 12

$$V_i P_i = R T_i,$$

and $$V_f P_f = R T_f,$$

but as $$V_i = V_f,$$

hence, $$P_f = \frac{T_f}{T_i} P_i. \quad . \quad . \quad . \quad . \quad . \quad (26)$$

Changes in the Temperature of One Pound of Air, at Constant Pressure.—The heat required to increase the temperature of one pound of air, at a constant pressure, is

$$Q = c_p (t_f - t_i).$$

When the temperature decreases the heat is rejected.

At a rising temperature the air performs the work

$$L = P_i (V_f - V_i) = R (t_f - t_i), \quad . \quad . \quad . \quad (27)$$

and at a decreasing temperature the same work is absorbed.

Inserting, successively, in equation 12, the initial volume and the final volume of the air, we obtain the ratio

$$\frac{V_f}{V_i} = \frac{T_f}{T_i}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

Isothermal Expansion and Compression.—When the temperature is constant equation 24 becomes

$$\phi_f - \phi_i = (c_p - c_v) \, log_e \frac{V_f}{V_i}.$$

Thus when air expands, the heat supplied is

$$Q = (c_p - c_v) \, T_i \, log_e \frac{V_f}{V_i}, \quad . \quad . \quad . \quad . \quad (29)$$

but, according to equation 14,

$$c_p - c_v = \frac{1}{J} R$$

hence,

$$Q = \frac{1}{J} R \, T_i \, log_e \frac{V_f}{V_i} = \frac{1}{J} V_i P_i \, log_e \frac{V_f}{V_i}. \quad . \quad (29a)$$

When air is being compressed, it rejects the heat

$$Q = (c_p - c_v) \, T_i \, log_e \frac{V_i}{V_f}, \quad . \quad . \quad . \quad . \quad (30)$$

or

$$Q = \frac{1}{J} R \, T_i \, log_e \frac{V_i}{V_f} = \frac{1}{J} V_i P_i \, log_e \frac{V_i}{V_f}. \quad . \quad (30a)$$

When expanding, it performs the work

$$L = R \, T_i \, log_e \frac{V_f}{V_i} = V_i P_i \, log_e \frac{V_f}{V_i}, \quad . \quad . \quad . \quad (31)$$

and for its compression is required the work

$$L = R \, T_i \, log_e \frac{V_i}{V_f} = V_i P_i \, log_e \frac{V_i}{V_f}. \quad . \quad . \quad (31a)$$

The temperature being constant, we obtain from equation 12

$$V_i P_i = V_f P_f,$$

an equation which expresses the law of Mariotte, and which can be written, in a general way,

$$V\ P = K;$$

K signifying a constant quantity.

Written in this form, the expression is the equation for the isothermal line $B\,C$, Fig. 1.

The area $A\,B\,C\,D$ underneath the isothermal line is, according to equation 2,

$$F = K \log_e \frac{V_f}{V_i},$$

but as $$K = V_i P_i,$$

hence, $$L = F = V_i P_i \log_e \frac{V_f}{V_i} \qquad (32)$$

From this equation may be obtained the common expression for the mean effective pressure in a steam-engine cylinder, when the expansion is assumed to be according to the isothermal line.

The work of expansion, given by equation 32, is, in Fig. 6,

FIG. 6.

represented by the area $C\,D\,E\,F$. Adding to this area the area $A\,B\,C\,F = V_i P_i$, representing the work of admission, we get the area $A\,B\,C\,D\,E$, which, therefore, represents the total work obtained during one stroke of the piston. Thus:

$$\therefore \quad L_t = V_i P_i + V_i P_i \log_e \frac{V_f}{V_i}.$$

The area representing the total work, divided by its length V_f, gives the mean pressure during the stroke.

Hence,
$$P_m = \frac{P_i \left(1 + \log_e \frac{V_f}{V_i}\right)}{\frac{V_f}{V_i}}.$$

If the expansion ratio $\frac{V_f}{V_i}$ be denoted by r, then the equation becomes

$$P_m = \frac{P_i (1 + \log_e r)}{r}.$$

Adiabatic Expansion and Compression—(No heat lost or received).—The change in the entropy during an adiabatic transformation being zero, equation 23 becomes

$$c_p \log_e \frac{V_f}{V_i} = c_v \log_e \frac{P_i}{P_f}.$$

From this, when according to notations adopted $\frac{c_p}{c_v} = n$, we obtain

$$\frac{P_i}{P_f} = \left(\frac{V_f}{V_i}\right)^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

The equation may be written

$$P_i V_i V_i^{n-1} = P_f V_f V_f^{n-1},$$

and, hence,
$$\frac{P_i V_i}{P_f V_f} = \left(\frac{V_f}{V_i}\right)^{n-1} = \left(\frac{P_i}{P_f}\right)^{\frac{n-1}{n}} = \frac{T_i}{T_f} \quad . \quad . \quad . \quad (33a)$$

When air expands its temperature declines, and it performs, according to equation 16, the work

$$L = J c_v (T_i - T_f),$$

and as, according to equations 12 and 14, we may write

$$R = \frac{V_i P_i}{T_i} = J (c_p - c_v) = J c_v (n - 1),$$

therefore, *when air expands it performs the work*

$$L = \frac{V_i P_i}{n-1}\left(1 - \frac{T_f}{T_i}\right),$$

or more conveniently $L = \frac{V_i P_i}{n-1}\left[1 - \left(\frac{V_i}{V_f}\right)^{n-1}\right]$ . . . (34)

Again, through substitution, $L = \frac{V_i P_i}{n-1}\left[1 - \left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}}\right]$. (35)

This work is, in Fig. 6, represented by the area $C\ D\ E\ F$, when $C\ D$ is an adiabatic line. The area $A\ B\ C\ F$ represents the work of admission to a cylinder.

Adding together the areas $A\ B\ C\ F$ and $C\ D\ E\ F$ we get *the total work L_t of admission and expansion*, represented by the area $A\ B\ C\ D\ E$. Thus

$$L_t = V_i P_i + \frac{V_i P_i}{n-1}\left[1 - \left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}}\right]$$

$$\therefore L_t = \frac{V_i P_i}{n-1}\left[n - \left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}}\right],$$

or $L_t = \frac{V_i P_i}{n-1}\left[n - \left(\frac{V_i}{V_f}\right)^{n-1}\right]$ . . . . . (36)

If P_3 be the back-pressure, then *the effective work of admission and expansion*, L_e, represented by the area $H\ B\ C\ D\ G$, becomes

$$L_e = L_t - P_3 V_f.$$

The mean pressure during the stroke is

$$P_m = \frac{V_i P_i}{(n-1)\ V_f}\left[n - \left(\frac{V_i}{V_f}\right)^{n-1}\right],$$

or $P_m = \frac{P_i}{n-1}\left(\frac{n}{r} - \frac{1}{r^n}\right)$ . . . . (37)

if the expansion ratio $\frac{V_f}{V_i}$ be denoted by r.

The mean effective pressure is

$$M.E.P = P_m - P_3.$$

If the air be assumed to expand completely down to the back-pressure, thus $P_f = P_3$, then the effective work of admission and expansion becomes

$L_e = \frac{n}{n-1} V_i P_i \left[1 - \left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}}\right]$ . . (38)

The work required to compress air is the same as that which it performs when expanding. P_i and V_i being the initial pressure and volume of the air, we get

$$L = \frac{V_i P_i}{n-1}\left[\left(\frac{V_i}{V_f}\right)^{n-1} - 1\right], \quad . \quad . \quad . \quad (34a)$$

$$\text{or } L = \frac{V_i P_i}{n-1}\left[\left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}} - 1\right] \quad . \quad . \quad . \quad (35a)$$

The work required to deliver the compressed air into a receiver, against the pressure P_f, is

$$V_f P_f,$$

and the work done by the outside atmosphere acting on the back of the compressor piston is

$$V_i P_i.$$

Thus, *the net work required to compress and deliver the air into a receiver* is

$$L_e = V_f P_f - V_i P_i + \frac{V_i P_i}{n-1}\left[\left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}} - 1\right].$$

From equation 33 we get

$$V_f P_f - V_i P_i = V_i P_i\left[\left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}} - 1\right],$$

which substituted in the expression for L_e reduces it to

$$L_e = \frac{n}{n-1} V_i P_i\left[\left(\frac{P_f}{P_i}\right)^{\frac{n-1}{n}} - 1\right]. \quad . \quad . \quad (39)$$

This work expressed in terms of the volume and pressure of the compressed air becomes

$$L_e = \frac{n}{n-1} V_f P_f\left[1 - \left(\frac{P_i}{P_f}\right)^{\frac{n-1}{n}}\right]. \quad . \quad . \quad (40)$$

CHAPTER II

DESIGN CONSTANTS AND FORMULAS

Principal Gas Power-Cycles.—Before entering into the immediate subject outlined for the present chapter, it may be desirable to present, in advance, a brief elucidation of the principal power-cycles which have been made a basis for the heat-transformation in the gas-engine, and in this connection it is not without interest to follow, in their historical order, the three fundamental inventions made during the years 1860 to 1873, from which the modern gas-engine practice is the logical development.

The leading inventor in this field, Lenoir, evidently took the steam-engine as a model, and through modifications of details changed the steam-engine into an internal-combustion engine. The engine he designed is shown by a sectional plan-view, in Fig. 7, and the result to which he arrived is represented by the diagram he obtained, in Fig. 8.

The engine draws its charge of gas and air of atmospheric pressure, during the first part of the stroke, and at a suitable point, one-third to one-half of the stroke, the inlet valve is closed and the charge ignited; the combustion, then, causing the pressure to rise suddenly, from that of the atmosphere to about 50 to 60 pounds above the atmosphere. During the remaining part of the stroke the expansion of the working charge takes place, and is continued until the atmospheric pressure is reached, at the end of the stroke. During the return stroke the charge is exhausted. in front of the piston, while the combined suction and expansion stroke is repeated in the end of the cylinder back of the piston. During the entire working cycle, there will be obtained, thus, for the head-end of the cylinder, the diagram *a b c d; a b* indicating the suction line, *b c* the combustion line, *c d* the expansion line and *d a* the exhaust line. For the crank-end of the cylinder a similar, reversed, card will be obtained.

The first important improvement upon this engine was suggested by Beau de Rochas, and effected by Otto, and from it there has gradually developed the modern Otto engine represented in Fig. 9. The main difference between the Lenoir and the Otto

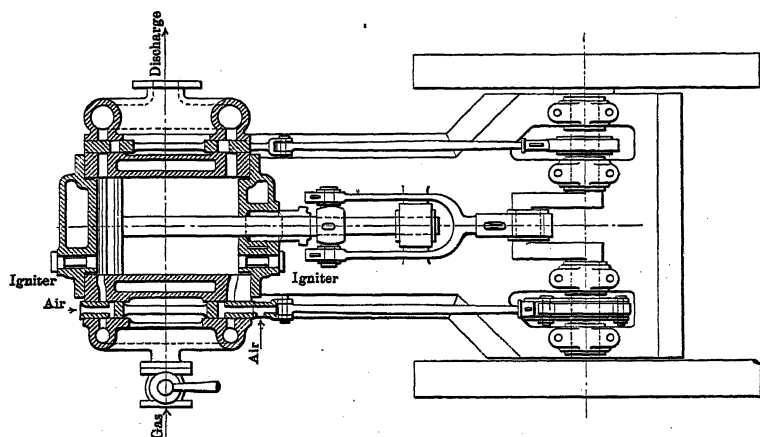



FIG. 7.—Sectional Plan. Lenoir Engine.

cycle is that the explosive charge is in the latter compressed to a pressure far above the atmospheric before it is ignited, by which is effected a most material increase in the initial explosion

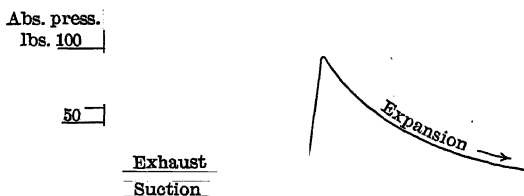



FIG. 8.—Indicator Card from Lenoir Engine.

pressure, as well as in the mean effective pressure during the working stroke.

Otto decided that in order to obtain two impulses for each revolution as the Lenoir engine gave, or as the steam-engine

would give, he had to employ four cylinders; and he built, accordingly, his four-cylinder engine. The Otto four-cycle engine most generally known to-day is, however, a one-cylinder engine giving one impulse every fourth stroke only, or one impulse every second revolution. In Fig. 9, the upper valve is the inlet valve, through which the explosive mixture of gas and air, in a suitable proportion, is admitted to the cylinder, and the lower, the exhaust valve, discharges the gases after their potential heat-energy has, through combustion, increase of pressure, and expansion, been duly converted into work. Each one of the

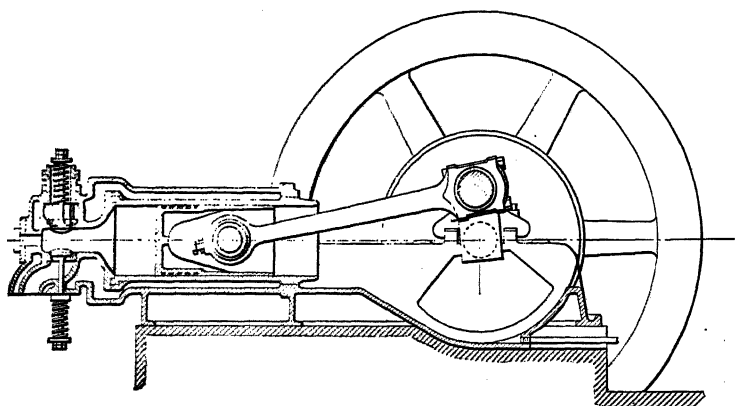



FIG. 9.—Sectional Elevation. Otto Engine.

valves being open only once during each two revolutions of the engine, they are actuated, generally, from a secondary shaft running along the side of the engine and made to revolve only one-half the number of turns of the main engine shaft.

The Otto cycle, shown graphically in Fig. 10, comprises five events, which are approximately represented by the lines traced on the indicator card of the modern engine. They are: The suction line, the compression line, the combustion line, the expansion line and the exhaust line. These lines represent, each, a separate stroke of the engine, excepting the combustion line, which indicates the sudden increase in pressure due to the explosion of the compressed charge, at the moment when the piston

is reversing from the return compression stroke to the forward expansion or working stroke. The suction line falls slightly below the line of the atmospheric pressure, due to the slight vacuum created by the suction of the piston, while the exhaust line falls slightly above the atmospheric line, due to the slight pressure created in displacing the exhaust gases from the cylinder.

It is during the expansion or working stroke, only, that useful work is being done by the gases, which press on the piston-head

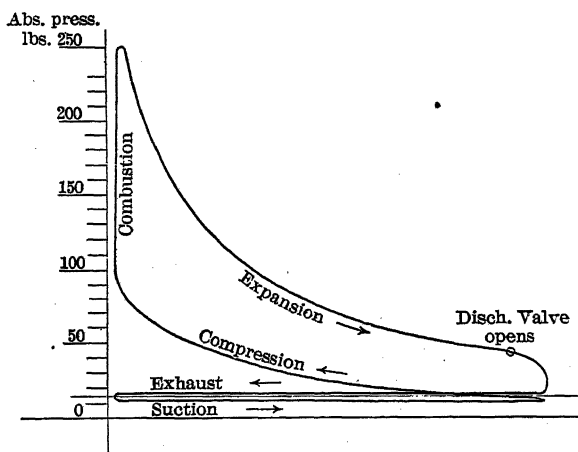



Fig. 10.—Indicator Card from Otto Engine.

with a gradually decreasing force. During the compression stroke, on the contrary, work must be consumed in forcing the piston against the increasing pressure of the gases undergoing compression.

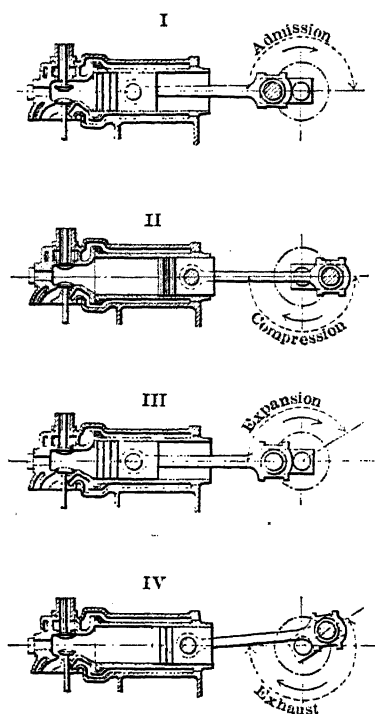
In order to convey to the reader as clear a view as possible of the different events, in the order they occur during the four strokes of the cycle, the four small sections of the engine, Fig. 11, are submitted; each figure showing plainly, by the position of the valves and piston, and by the direction of motion of the latter, the function performed by the engine during each stroke.

With the development of the Otto engine, it was soon discovered that approximately the same cycle as the one just de-

scribed could well be obtained during only two strokes of the piston, simply by forcing in to the cylinder the new charge, and thereby replacing the old one, during the interval when the piston is reversing after a completed expansion stroke. This idea of completing the cycle during two strokes of the piston, instead of using four as the Otto engine required, led to the Clerk, the Bentz, the Koerting, and the Oechelhäuser engines—in short, to the development of the modern two-cycle engine.

FIG. 11.

A third gas-engine cycle, which is most important with respect to certain kinds of fuel, was originated by Brayton, and is represented in the modern Diesel engine.

A sectional view of the cylinder of this engine, showing plainly the general arrangement of its valves, is given in Fig. 12. The fuel employed is oil, and it is admitted at a high pressure through the fuel-admission valve, marked "fuel needle-valve" in the figure. *A* is the air-admission valve and *D* the discharge valve for the waste gases. For a more detailed description of the general construction of the engine see page 387.

This power-cycle is in practice carried out through four strokes of the engine, which operates as follows:

Pure air is drawn in to the cylinder during the downward suction stroke, and compressed into the combustion-chamber, during the return stroke of the piston, to a very high pressure. The fuel-oil is then gradually introduced by means of an oil-pump, to an amount determined by the position of the governor,

and burned during the first part of the following pressure stroke, until, at a certain point, the fuel is cut off and expansion takes place for the rest of the stroke. The fuel is consumed at such a rate as to maintain the maximum temperature and pressure of the working charge approximately constant during the period

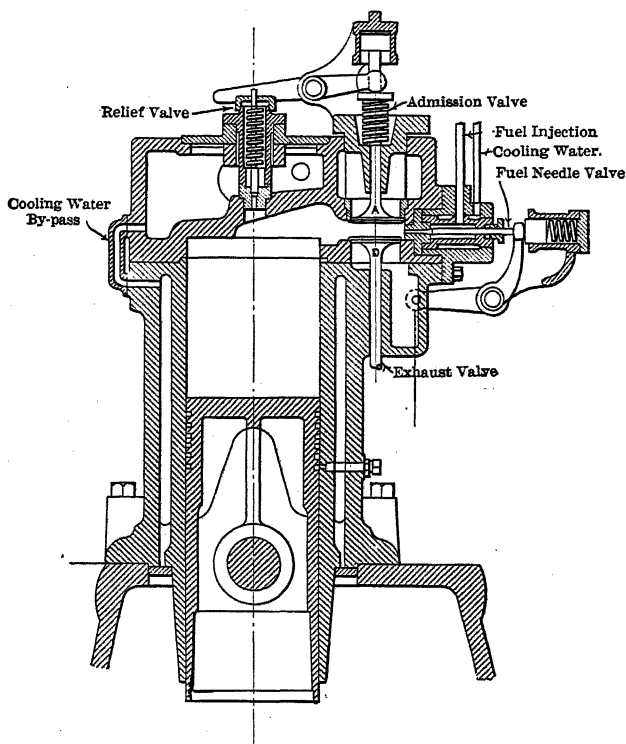



FIG. 12.—Section through Diesel Cylinder

of combustion. The diagram representing the cycle, Fig. 13, will, therefore, show a nearly constant pressure line, *a b*, during the first part of the working stroke. At the point *b* the fuel-charge is cut off, and the expansion line continues from there to the end of the stroke. The suction, compression, and exhaust lines are identical with those of the Otto diagram.

The Normal Cycle.—In connection with the design of engines which are to work under stipulated conditions, certain questions,

as to the maximum pressure, the mean effective pressure, the efficiency of a contemplated heat-transformation, etc., are constantly recurring. It becomes, therefore, of interest to adopt a set of formulas, by means of which may be forecasted the results

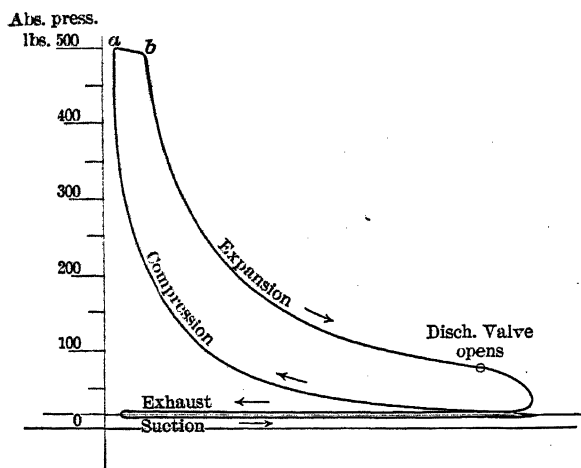
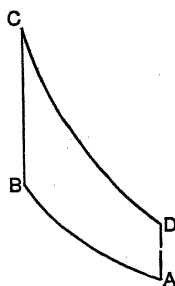



FIG. 13.—Indicator Card from Diesel Engine.

that can be expected under normal working conditions, and the derivation of such formulas, directly applicable to the gas-engine cycle, will be the object in the present chapter.

FIG. 14.

It is to be noted that there are to be found in connection with the performance of the gas-engine, a number of factors that are of what may be called an accidental nature, each of which may influence the results of individual cycles, but which are foreign to the normal cycle. For instance, among a number of cards, taken on a constant load, no two cards may be found exactly alike, some showing a decidedly higher initial pressure, mean pressure, or final pressure than others. A formula can on this

account only approximate the mean results obtained during a number of cycles, or the results "under normal working conditions."

Pressure and Temperature of the Compressed Charge.—The Otto cycle which forms the basis for the derivations in the following is represented by the pressure-volume diagram Fig. 14. Its characteristics are: The compression line $A\ B$ and expansion line $C\ D$ are adiabatics, and the heat is received from B to C and rejected from D to A at constant volumes.

Let T designate absolute temperatures,
P absolute pressures, in pounds per square foot,
V volumes,

and let the indices a, b, c, and d designate the points where the events of the cycle change from one into another, according to Fig. 14.

Thus T_a = temperature before compression
T_b = temperature after compression
T_c = temperature after combustion
T_d = temperature after expansion

and similarly in respect to pressures and volumes.

Due to the assumed adiabatic compression of the charge, we have according to equations 33 and 33*a*

$$P_b = \left(\frac{V_a}{V_b}\right)^n P_a = r^n P_a,$$

$$T_b = \left(\frac{V_a}{V_b}\right)^{n-1} T_a = r^{n-1} T_a, \quad . \quad . \quad . \quad (33a)$$

when for convenience $\frac{V_a}{V_b}$ is written r, and $\frac{c_p}{c_v} = n$; c_p being the specific heat at constant pressure and c_v the specific heat at constant volume of the explosive mixture.

The specific heat is not the same for all gases. Nor does it seem likely, in the face of determinations indicating the contrary, that the specific heat is the same at high temperatures as at low. Within the range of temperatures at which the compression in the gas-engine takes place, there is not any great error involved, however, in considering c_p and c_v constants, but the specific heats of the expanding gases, which are of a different nature and tem-

perature than those subject to compression, cannot be assumed to be the same as the values c_p and c_v, unless a proper correction is made for the error introduced by such an assumption.

Pressure and Temperature after Combustion.—The heat supplied to a working charge is generally designated by the letter Q. In speaking of a theoretical cycle it is understood, that if Q heat-units have been supplied to a working charge Q heat-units have also been absorbed by it as internal or external heat. But stating that Q heat-units have been supplied to the working charge in a gas-engine cylinder does not necessarily signify that Q heat-units have been absorbed by it. On the contrary, when the calorific heat of a charge is set free, at the end of the compression stroke, the gases can absorb only a part of it, because partly it must necessarily be absorbed by the metallic surfaces composing the combustion-chamber, or it may otherwise be dissipated or suppressed.

As the following will have reference to the practical gas-engine cycle, as far as possible, distinction must be made between the heat supplied and the heat absorbed by the working gases; and there arises, at the outset, a correction factor, f, for the correction for the heat apparently lost at the combustion, on the assumption that c_v is constant.

Let, therefore, Q designate the heating-value supplied per pound of *normal charge*, when by normal charge is meant the gas and air-mixture, together with the proportion of neutrals remaining in the combusion-chamber from the preceding stroke;

fQ the heating-value absorbed by the working gases at the time of combustion, and accounted for by the rise in temperature expressed by the formula

$$T_c = \frac{fQ}{c_v} + T_b.$$

Thus, f may be defined as the ratio of the heat-value theoretically absorbed by the working gases to the total heat supplied in the normal charge.

By transposition of the formula above we obtain

$$T_c = T_b \left(1 + \frac{fQ}{c_v T_b}\right). \quad . \quad . \quad . \quad . \quad (41)$$

But, as the volume remains constant during the combustion, thus

$$\frac{P_c}{P_b} = \frac{T_c}{T_b},$$

hence

$$P_c = P_b \left(1 + \frac{fQ}{c_v T_b} \right). \quad . \quad . \quad . \quad . \quad (42)$$

The ratio of the maximum pressure to the compression pressure will be

$$\frac{P_c}{P_b} = 1 + \frac{fQ}{c_v T_b}$$

or

$$\frac{P_c}{P_b} = 1 + \frac{fQ}{c_v T_a r^{n-1}}. \quad . \quad . \quad . \quad . \quad (43)$$

Heating-value of the Charge.—The heating-value of a fuel-gas is generally given in heat-units per cubic foot of the gas, at a standard pressure and temperature. That is, at a pressure of one atmosphere and at a temperature of 62° F. Let this heating-value be H, and assume that the volume of air required for a theoretically perfect combustion of each cubic foot of gas, according to chemical analysis, is a known quantity and that it is "a" cubic foot.

The volume of air required for a perfect combustion in the gas-engine cylinder would not merely be the volume, a, required according to chemical analysis, but it must include a proper margin in excess, say, 15 per cent or whatever experience has found to be most efficient. Let this volume be called $x\,a$.

The heating-value in each cubic foot of a practically perfect gas-and-air mixture is, then, $\frac{H}{x\,a + 1}$ B.T.U., at a standard pressure and temperature.

Assuming H_a to be the heating-value of one cubic foot of fuel-gas, at the pressure P_a, absolute temperature T_a, and specific volume V_a, then we may write

$$H_a = \frac{V_o}{V_a} H;$$

the heating-value per cubic foot of gas changing in the inverse ratio to its specific volume. V_o is assumed to be the specific volume of standard gas.

Should the pressure of the gas be standard, but not its temperature, then its heating-value, per cubic foot, would vary inversely as its absolute temperature, and it becomes

$$H_a = \frac{T_o}{T_a} H.$$

The heating-value per cubic foot of a perfect mixture is, therefore,

$$\frac{H_a}{x a + 1} = \frac{V_o}{V_a} \frac{H}{x a + 1},$$

or

$$\frac{H_a}{x a + 1} = \frac{T_o}{T_a} \frac{H}{x a + 1}, \text{ at the pressure } P_a.$$

Under normal working conditions $x a$ would be the volume of air which is required, per cubic foot of gas, for the most effective combustion; however, with respect to an engine under test, $x a$ must, of course, denote the actual quantity of air used.

The factor $\dfrac{H}{\frac{V_a}{V_o}(x a + 1)}$ becomes in the latter case the heating-value per cubic foot of suction-displacement.

The Normal Charge—Increase in Pressure through Combustion.—The actual charge in a non-scavenging gas-engine cylinder is not composed of fuel-gas and air only, but it contains neutral gases, to the amount of the volume of the clearance-space.

The volume of the fresh mixture drawn in by the piston is

$$V_a - V_b,$$

and the corresponding volume of the combustion-chamber is

$$V_b,$$

therefore, the volume of neutral gases mixed with $(x a + 1)$ cubic foot of fresh mixture must be

$$N = \frac{V_b}{V_a - V_b}(x a + 1) = \frac{1}{r - 1}(x a + 1),$$

but since the volume of the normal charge resulting from one cubic foot of fuel-gas is

$$N + a x + 1,$$

hence, the heating-value per cubic foot of normal charge becomes

$$\frac{Q}{V_a} = \frac{H_a}{N + (x a + 1)} = \frac{H_a (r - 1)}{(a x + 1) r} \text{ B.T.U.},$$

and

$$Q = \frac{H_a V_a (r - 1)}{(x a + 1) r} \text{ B.T.U.}$$

But as $H_a = \frac{V_o}{V_a} H,$

therefore,

$$Q = \frac{H}{x a + 1} \frac{V_o (r - 1)}{r} = \frac{H V_a}{\frac{V_a}{V_o} (x a + 1)} \frac{(r - 1)}{r} \quad . \quad (44)$$

Substituting this value for the heat supplied in the equation for the maximum pressure (43), it becomes

$$P_c - P_b = \frac{f P_a V_a (r - 1)}{c_v T_a} \frac{H}{\frac{V_a}{V_o} (x a + 1)} \quad . \quad . \quad (45)$$

The fundamental equations for perfect gases are:

$$R = \frac{V_a P_a}{T_a}, \quad \text{and} \quad R = J (c_p - c_v).$$

Hence,

$$\frac{V_a P_a}{T_a} = J \cdot c_v (n - 1).$$

This value substituted in the equation 45 gives

$$P_c - P_b = J f (r - 1) (n - 1) \frac{H}{\frac{V_a}{V_o} (x a + 1)} \quad . \quad (45a)$$

Through division by 144 is obtained the maximum increase in pressure, expressed in pounds per square inch, thus

$$p_c - p_b = 5.4 f (r - 1) (n - 1) \frac{H}{\frac{V_a}{V_o} (x a + 1)}, \quad . \quad (45b)$$

or if $n = 1.35$

$$p_c - p_b = 1.89 f (r - 1) \frac{H}{\frac{V_a}{V_o} (x a + 1)} \quad . \quad . \quad (45c)$$

When applying the preceding formulas with respect to actual

indicator cards, r must be considered to represent the actual compression ratio corresponding to the pressures P_a and P_b; that is, $r = \left(\frac{P_b}{P_a}\right)^{\frac{1}{n}}$. On account of practical conditions the ratio $\frac{V_a}{V_b}$ will be less reliable.

The assumption is, under these conditions, that the neutrals and fuel mixture occur in the final charge in the ratio $\left(\frac{P_b}{P_a}\right)^{\frac{1}{n}}$.

The equations for the coefficient f will be

$$f = \frac{1}{5.4} \frac{p_c - p_b}{(r-1)(n-1)} \frac{1}{\dfrac{H}{\dfrac{V_a}{V_o}(xa+1)}}, \quad (46)$$

and, according to equation 42,

$$f = \frac{\left(\dfrac{p_c}{p_b} - 1\right) c_v T_a r^{n-1}}{Q}. \quad (46a)$$

The following expressions may also be useful:

$$f = \frac{1}{5.4} \frac{p_c - p_b}{(r-1)(n-1)} \frac{\text{Suct. displ. per I.H.P. per min.}}{\text{H.U. cons. per I.H.P. per min.}} \quad (46b)$$

$$f = \frac{1}{5.4} \frac{p_c - p_b}{(r-1)(n-1)} \frac{\text{Suction displ. per min.}}{\text{H.U. cons. per min.}} \quad . \quad . \quad (46c)$$

Work Generated, Theoretical Efficiency.—Let, as before, Q designate the total heating-value supplied per pound of the normal charge; fQ the heating-value absorbed by the charge at the time of combustion; and Q_2 the heating-value that has been rejected at a constant volume, at the time of release.

Thus

$$fQ = c_v(T_c - T_b) = c_v T_b \left(\frac{T_c}{T_b} - 1\right),$$

$$Q_2 = c_v(T_d - T_a) = c_v T_a \left(\frac{T_d}{T_a} - 1\right).$$

and

$$\frac{fQ}{Q_2} = \frac{T_b}{T_a} \frac{\dfrac{T_c}{T_b} - 1}{\dfrac{T_d}{T_a} - 1}.$$

If the compression as well as the expansion of the charge be assumed to be adiabatic, we may write:

$$\frac{T_b}{T_a} = \left(\frac{V_a}{V_b}\right)^{n-1}$$

$$T_c = \left(\frac{V_d}{V_c}\right)^{n-1} T_d,$$

and $$T_b = \left(\frac{V_a}{V_b}\right)^{n-1} T_a.$$

But as $V_a = V_d$, $V_b = V_c$ and $\frac{V_a}{V_b} = \frac{V_d}{V_c}$,

therefore $\frac{T_c}{T_b} = \frac{T_d}{T_a}$,

and hence $\frac{fQ}{Q_2} = \frac{T_b}{T_a} = \left(\frac{V_a}{V_b}\right)^{n-1} = r^{n-1}$; if, as before, $\frac{V_a}{V_b}$ be designated by r.

The theoretical efficiency of the cycle, which may be expressed as the ratio between the heat converted into work and the heat absorbed initially by the working gases, becomes

$$E = \frac{fQ - Q_2}{fQ} = 1 - \frac{1}{r^{n-1}}, \quad . \quad . \quad . \quad (47)$$

and the work performed is

$$L = J\,(fQ - Q_2) = J\,E\,fQ = J\left(1 - \frac{1}{r^{n-1}}\right) fQ. \quad (48)$$

The practical cycle, such as carried out in the gas-engine cylinder, should give approximately this result, if both the compression and expansion-lines were adiabatics, and if the addition and discharge of heat were effected at constant volumes. But heat may have been gained or lost during the practical cycle, and, hence, a correction of the efficiency E is required.

Assume the efficiency of the practical cycle to be $yE = y\left(1 - \frac{1}{r^{n-1}}\right)$, then the work performed per pound gas becomes

$$L = J\,E\,f\,y\,Q = J\left(1 - \frac{1}{r^{n-1}}\right) f\,y\,Q. \quad . \quad (48a)$$

Thermal Efficiency.—The factor Efy is the percentage of the heat-energy supplied that becomes transformed into work,

and it is commonly referred to as the thermal efficiency of the heat-transformation. This quantity varies considerably for different engines and fuels, but under favorable conditions the values given in Table I should be approached.

The coefficient $E = 1 - \frac{1}{r^{n-1}}$ is a constant quantity for all engines of the same ratio of compression, and its value, computed for several compression ratios within practical limits, will be found in Table I. The same table contains, for convenience in ordinary computation, also the values of r^n, r^{n-1}, $\frac{1}{r^{n-1}}$, for n 1.35.

TABLE I.

Average Efficiency for Various Compressions

Ratio of Compression.	r^n	r^{n-1}	$\frac{1}{r^{n-1}}$	E $1 - \frac{1}{r^{n-1}}$	Efy
3	4.41	1.47	0.680	0.320	0.190
3.5	5.44	1.55	0.646	0.354	0.220
4	6.50	1.62	0.620	0.380	0.240
4.5	7.60	1.69	0.596	0.404	0.255
5	8.76	1.75	0.574	0.426	0.268
5.5	9.96	1.81	0.553	0.447	0.280
6	11.22	1.87	0.534	0.466	0.290
6.5	12.51	1.93	0.518	0.482	0.300
7	13.83	1.98	0.504	0.496	0.305
7.5	15.19	2.03	0.493	0.507	0.310

The Mean Effective Pressure.—If A designates the area of the piston, in square feet, and Z the total length of the suction-strokes that correspond to the admission of one pound of charge, then

$$A Z = V_a - V_b$$

$$= V_a \left(1 - \frac{V_b}{V_a}\right) = V_a \left(1 - \frac{1}{r}\right),$$

and the theoretical mean effective pressure, expressed in pounds per square foot,

$$M.E.P. = \frac{L}{A\,Z} = J\,\frac{E\,f\,y}{1 - \frac{1}{r}}\,\frac{Q}{V_a} \tag{50}$$

It has been shown that

$$Q = \frac{H}{x\,a + 1}\,V_o\,(r - 1) \quad \text{equation 44,}$$

hence, $$M.E.P. = J\,E\,f\,y\,\frac{H}{\frac{V_a}{V_o}\,(x\,a + 1)} \quad \text{pounds per sq. ft.} \tag{50a}$$

J being 778, and there being 144 square inches per square foot area, the mean effective pressure, expressed in pounds per square inch, will be

$$m.e.p. = 5.4\,E\,f\,y\,\frac{H}{\frac{V_a}{V_o}\,(x\,a + 1)} \quad . \quad . \quad . \tag{50b}$$

E is the theoretical efficiency of the cycle,
f represents the fraction by which the heat-value supplied in the charge has become useful in raising the pressure during the initial combustion, and
y a correction-factor for the heat that has been gained or lost during the practical cycle.

$\dfrac{H}{\frac{V_a}{V_o}\,(x\,a + 1)}$ is the heating-value per cubic foot of suction displacement.

The factor E increasing with increased compression, it will be evident from the above equation, that the mean effective pressure will become higher when a higher compression is used, provided the factor $f\,y$ does not diminish thereby, to offset the increase gained.

A diagram such as $A\,B\,C\,D$, Fig. 14a, which is assumed to have been obtained by the adiabatic expansion of a perfect gas between the points C to D and its adiabatic compression between the points A and B, is often spoken of as a theoretical air-card.

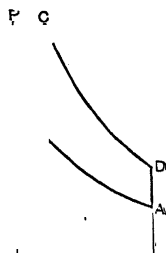



FIG. 14a.

The mean pressure of such a card is readily obtained from formulas 34 and 34*a*.

The work generated during the expansion C to D, the area $C\,D\,F\,E$, is

$$\text{(equation 34)} \qquad L_1 = \frac{V_c P_c}{n-1}\left[1 - \left(\frac{V_c}{V_d}\right)^{n-1}\right],$$

and the work absorbed during the compression A to B, the area $A\,B\,E\,F$, is

$$\text{(equation 34a)} \qquad L_2 = \frac{V_a P_a}{n-1}\left[\left(\frac{V_a}{V_b}\right)^{n-1} - 1\right],$$

or

$$= \frac{V_b P_b}{n-1}\left[1 - \left(\frac{V_b}{V_a}\right)^{n-1}\right].$$

The total work generated, the area $A\,B\,C\,D$, is, therefore,

$$L_1 - L_2 = V_b \frac{P_c - P_b}{n-1}\left[1 - \left(\frac{V_b}{V_a}\right)^{n-1}\right];$$

remembering that $V_b = V_c$ and $V_a = V_d$.

Hence, the M.E.P. of the air-card =

$$\frac{L_1 - L_2}{V_a - V_b} = \frac{V_b}{V_a - V_b}\,\frac{\left(1 - \frac{1}{r^{n-1}}\right)}{n-1}(P_c - P_b) = \frac{E\,(P_c - P_b)}{(r-1)\,(n-1)} \qquad (49)$$

The same formula may readily be obtained also by means of formulas 45*a* and 50*a*, assuming that $y = 1$.

Horse-power, Mechanical Efficiency.—The work generated from the heating-value of the unit weight of gas is in equation 48*a* expressed in foot-pounds. When the supply of heat-energy for transformation is continued at such a rate that 33,000 foot-pounds of work is obtained per minute, then the power generated by the engine is said to be one horse-power.

The work corresponding to one horse-power or 33,000 foot-pounds per minute may be expressed by its equivalent in heat-units, and it becomes often convenient to refer to the following relations:

One horse-power = 42.42 B.T.U. per minute.

One horse-power = 2,545 B.T.U. per hour.

The horse-power of an engine is commonly expressed in two

ways: as the power exerted on the piston—the indicated horse-power, or as the power that can be taken off from the engine-shaft by means of a brake and brake-wheel—the brake horse-power. The difference between the indicated power and the brake-power is the power lost in friction in the engine itself, and the percentage of the indicated power that will be obtained as brake-power is called the mechanical efficiency of the engine; hence,

$$\text{Mechanical efficiency} = \frac{B.H.P.}{I.H.P.}$$

Values of the Thermal Efficiency and m.e.p.—The coefficient Efy is the thermal efficiency of an engine, with reference to its indicated horse-power. In many gas-engine tests the thermal efficiency is counted inclusively of the mechanical efficiency of the engine, that is

$$\text{Thermal efficiency} = \frac{\text{Output in Brake H.P.}}{\text{Total heat supplied.}}$$

By dividing, however, the thermal efficiency thus expressed by the coefficient for the mechanical efficiency, we get the equivalent of the factor Efy.

$$\text{Thus} \qquad Efy = \frac{\text{Thermal efficiency in B.H.P.}}{\text{Mechanical efficiency.}}$$

When the value of the mechanical efficiency is not expressly stated, a mean value,

Mechanical efficiency = 0.85, may be used.

The heat-units consumed per horse-power per hour, or per minute, are generally determined by engine tests, and based on the data thus obtained the efficiency becomes,

$$Efy = \frac{2{,}545}{\text{H.U. cons'd per I.H.P. per hour}}, \quad . \quad (51)$$

or

$$Efy = \frac{42.42}{\text{H.U. cons'd per I.H.P. per min.}} \quad . \quad (51a)$$

The following expressions may sometimes be found convenient

$$Efy = \frac{42.42 \times \text{I.H.P.}}{\text{H.U. cons'd per min.}} \quad . \quad . \quad . \quad (51b)$$

$$E f y = \frac{m.e.p.}{5.4 \dfrac{H}{\dfrac{V_a}{V_o}(x a + 1)}} \quad . \quad . \quad . \quad . \quad . \quad (51c)$$

The displacement swept through by the piston during the suction-strokes (= suction displacement), can readily be obtained, and it may become convenient to use the following expressions for the m.e.p.

$$m.e.p. = 5.4 \; E f y \frac{\text{H.U. cons'd per I.H.P. per min.}}{\text{Suct. displ. per I.H.P. per min.}} \quad (50c)$$

$$m.e.p. = \frac{229}{\text{Suct. displ. per I.H.P. per min.}} \quad . \quad (50d)$$

$$m.e.p. = \frac{229 \quad \text{I.H.P.}}{\text{Total Suct. displ. per min.}} \quad . \quad (50e)$$

The latter formula is useful for checking up the mean effective pressure obtained at tests, to suit the actual suction-displacement, and the indicated horse-power developed, within due approximation. This is often neglected, leading to contradictions in the general result.

The suction- and pressure-displacements are, of course, the same, and in a hit-or-miss engine only the actual suction- or pressure-strokes should be counted.

The heat-units consumed per indicated horse-power being known, and the factor E determined from the compression ratio, the factor $f y$ may be obtained from the expression

$$f y = \frac{2{,}545}{\text{E} \times \text{H.U. cons'd per I.H.P. per hour}} \quad . \quad . \quad (51d)$$

Required Piston-displacement per Horse-power.—Let S be the piston speed, in feet per minute, then the corresponding indicated horse-power of a four-cycle single-acting engine becomes

$$I.H.P. = \frac{M.E.P. \; A \; S}{33{,}000 \times 4},$$

or

$$I.H.P. = J E f y \frac{H}{\dfrac{V_a}{V_o}(x a + 1)} \; \frac{A S}{33{,}000 \times 4},$$

$\frac{A S}{4}$ being the piston-displacement during the pressure-strokes.

If D be the minimum suction-displacement required per indicated horse-power for the particular gas to which the heating-value $\frac{H}{\frac{V_a}{V_o}(x a + 1)}$ refers, we may write:

$$D = \frac{33{,}000}{778 \; E f y \; \frac{H}{\frac{V_a}{V_o}(x a + 1)}} = \frac{42.42}{E f y \; \frac{H}{\frac{V_a}{V_o}(x a + 1)}} \quad . \quad (52)$$

$\frac{H}{\frac{V_a}{V_o}(x a + 1)}$ is the maximum heating-value per cubic foot of the expanded normal charge, at the end of the suction-stroke, and this factor can readily be estimated, as will be shown later, when the calorific value and analysis of the gas are known.

The thermal efficiency, $E f y$, that should be expected under normal good conditions, for various compression-ratios, can be obtained from Table IV, page 68.

By equation 50*a* it appears, that the greater the volume of air, xa, drawn in with the mixture the less the resulting M.E.P. will be, or, in other words, the less the volume of the gas-mixture carrying a certain heating-value, the higher will the M.E.P. be. This, of course, is true only unto the limit where the air is just enough for a complete combustion of the fuel-gas.

Again, the mixture will be high in heating-value, hence, the horse-power a maximum only when V_a is small. By throttling the charge, or by supplying it hot to the engine, V_a becomes large and the power small. An engine will, therefore, give a maximum power only when V_a is a minimum, and when xa is the most suitable for giving a good combustion.

The equations 45*b* and 50*b* may be written, respectively,

$$\frac{H}{\frac{V_a}{V_o}(x a + 1)} = \frac{1}{5.4} \; \frac{p_c - p_b}{f(n - 1)\;(r - 1)},$$

and
$$\frac{H}{\frac{V_a}{V_o}(xa+1)} = \frac{1}{5 \cdot 4}\,\frac{(m.e.p.)}{Efy},$$

Hence
$$\frac{p_c - p_b}{f(n-1)(r-1)} = \frac{(m.e.p.)}{Efy},$$

or
$$y = \frac{(m.e.p.)(n-1)(r-1)}{(p_c - p_b)E} \quad . \quad . \quad . \quad (53)$$

This equation is simply the ratio between the area of the actual card and the theoretical air-card, and may, of course, be derived directly from equation 49.

CHAPTER III

THEORETICAL ANALYSES OF THE GAS-ENGINE CYCLES

The Entropy-Temperature Diagram.—The entropy and temperature of a substance can, we have seen, be represented graphically with reference to co-ordinates, in a manner similar to that in which pressures and volumes are represented in the volume-pressure diagram.

Fig. 15 is such an entropy-temperature diagram, in which the absolute zero may be assumed to be located at the line $O - \phi$ and the lower temperature-limit of the cycle at A. Absolute temperatures are represented by heights above the base line $O - \phi$, whereas entropy is represented by distances from the line $O\,T$. Assume that the diagram has reference to the changes in entropy and temperature called for by the Otto cycle. Its starting-point may be assumed at A, where the initial conditions of the substance are represented by the absolute temperature T_a and entropy ϕ_1, and from where its compression is effected adiabatically. There being, during the compression, no loss or gain of heat, the entropy remains unchanged, and the substance arrives to the point B of a pressure P_b, absolute temperature T_b and entropy ϕ_1.

From this point heat is added through combustion in such a manner, that for each elementary increment in entropy, $\delta\phi$, there is an elementary increase in the absolute temperature, $\delta\,T$; when the combustion is completed the entropy is ϕ_2 and the absolute temperature is T_c. The change in entropy and temperature during the combustion is thus represented by the line $B\,C$. From C expansion takes place, adiabatically, along the line of constant entropy $C\,C_o$, until a point D is reached, after which the change in the condition of the substance will be along the line $D\,A$ representing its cooling at a constant volume; the cycle finally closing at the point A.

The area of the diagram, *A B C D*, represents the work generated during the cycle, and it can, of course, be expressed by the product of the entropy $\phi_2 - \phi_1$ times the mean height of the diagram expressed in degrees Fahrenheit.

The entropy at any point x of the cycle may be determined when the pressure and volume existing at the point is known, as well as the pressure and volume at any other point of the cycle.

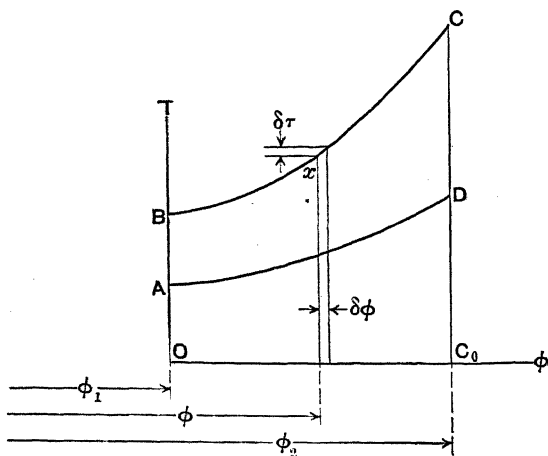



FIG. 15.

The initial pressure and volume (P_a, V_a) of the charge being the most convenient to refer to we may write, according to equation 23:

$$\phi_2 - \phi_1 = c_v \left(log_e \frac{P_x}{P_a} - \frac{c_p}{c_v} log_e \frac{V_a}{V_x} \right) \quad . \quad . \quad (54)$$

The scale in which $\phi_2 - \phi_1$ is laid off in the diagram being arbitrary, we may eliminate the factor c_v. Further, the ratio between the hyperbolic and common logarithms being a constant which will be cancelled at the division of the two logarithms, it does not matter which system of logarithms is used. Hence we may write

$$\phi_2 - \phi_1 = log \frac{P_x}{P_a} - 1.3 \, log \frac{V_a}{V_x}; \quad . \qquad (55)$$

$\frac{c_p}{c_v}$ being assumed 1.3.

The temperature at any point of the cycle is most conveniently determined by the equation

$$log \frac{T_x}{T_a} = log \frac{P_x}{P_a} - log \frac{V_a}{V_x}; \quad . \quad (56)$$

By solving these equations for different values P_x and V_x any number of points of the ϕ T-diagram may be obtained.

It is evident that, as the actual value of $\frac{c_p}{c_v}$ is not very definitely known, and as it probably changes materially for different points of the cycle, the entropy-temperature diagram will be

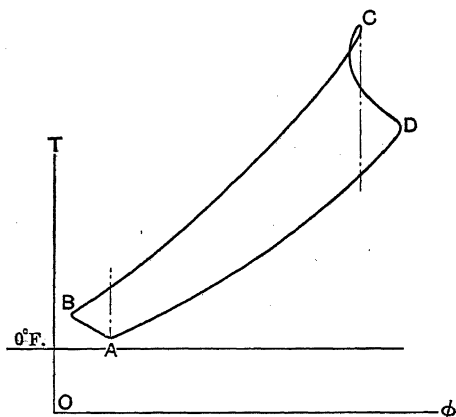



FIG. 16.

correct only in so far that it indicates the relative gain or loss of heat-energy from point to point of the cycle. Thus in Fig. 16, which is a ϕ T-diagram constructed from an actual indicator-card, Fig. 17, we notice that the expansion line C D, instead of being adiabatic as the theoretical cycle requires, shows that heat, to some extent, has been gained during the expansion. The compression line A B, on the contrary, shows loss of heat.

Principal Heat-Engine Cycles.—There are four principal cycles for the transformation of heat into work, besides a number of variations of these, of subordinate interest. Included in the four, the first one is only of historic interest, and will be considered in the following for the sake only of making the series complete.

The cycles consist, practically, of four or more events, according to the following table, although in the strictly theoretical cycle the admission or suction period is not considered; the same material being assumed to operate over and over.

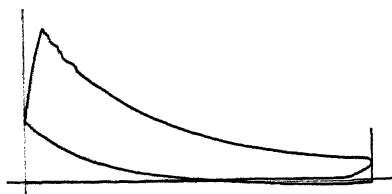



FIG. 17.

The indicator cards, Figs. 8, 10 and 13, represent the three first gas-engine cycles, I, II, and III, of Table II.

Comparison between Cycles.—Although there is, between the practical and theoretical gas-engine cycles, an approximate conformity only, it will, however, be of considerable interest to

TABLE II

Principal Cycles

	Admission or Suction.	Compression.	Heating.	Expansion.	Cooling.
Cycle I. Lenoir	At atmospheric pressure	None	At const. volume	Adiabatic	At const. pressure (isopiestic)
Cycle II. Otto		Adiabatic	At const. volume	Adiabatic	At const. volume (isometric)
Cycle III. Brayton		Adiabatic	At const. pressure	Adiabatic	At const. pressure (isopiestic)
Cycle IV. Carnot Ideal		Adiabatic	Isothermal	Adiabatic	Isothermal

compare the results that, due to theoretical considerations, must be expected of each type, because thereby the faults or advantages of any of them may readily be detected. Thus, a comparison between the cycles in respect to efficiency, temperature range, maximum pressure and effectiveness (the latter being judged by the mean effective pressure obtained), must be of first interest.

I.

In the Lenoir cycle the combustion is effected without previous compression, the charge simply being drawn into the cylinder,

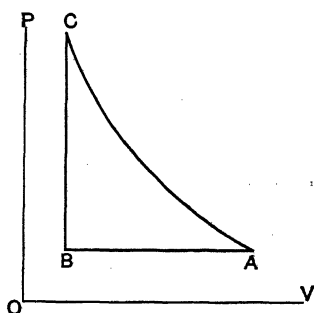
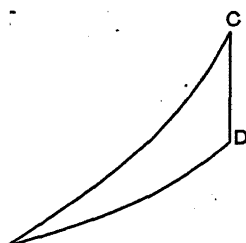



FIG. 18. FIG. 19.

during the first part of the stroke, and exploded at a point suitable to the load.

Figs. 18 and 19 represent, respectively, a volume-pressure and an entropy-temperature diagram of the cycle. The successive events are the following:

From B to C. The heating of the charge is effected at constant volume, from the atmospheric pressure.

From C to D. The working gases are expanded, adiabatically.

From D to B. They are expelled or cooled at atmospheric pressure.

Let T designate absolute temperatures, P pressures, in pounds per square foot, V volumes;
and let the indices b, c, and d designate the points where the

separate events of the cycle change from one into another, according to the figures.

Thus, T_b = temperature before combustion,
T_c = temperature after combustion,
T_d = temperature after expansion,
and similarly with respect to pressures and volumes.

c_p and c_v are the specific heats of the working substance, respectively, at constant pressure and at constant volume, and

$$c_v \quad \ldots$$

The heat transmitted to the charge during the period B to C is

$$Q_1 = c_v (T_c - T_b), \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

and the heat rejected from D to B is

$$Q_2 = c_p (T_d - T_b).$$

The efficiency of the transformation, therefore,

$$E = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} = 1 - n \frac{T_d - T_b}{T_c - T_b}.$$

$$\therefore E = 1 - n \frac{\frac{T_d}{T_b} - 1}{\frac{T_c}{T_b} - 1}.$$

During the period of constant pressure D to B the ratio between the temperatures is

$$\frac{T_d}{T_b} = \frac{V_d}{V_b},$$

and during the period of constant volume B to C

$$\frac{T_c}{T_b} = \frac{P_c}{P_b}.$$

According to equation 33 the ratio between the pressures during the adiabatic expansion C to D is

$$\frac{P_c}{P_d} = \left(\frac{V_d}{V_c}\right)^n,$$

But, as $P_d = P_b$ and $V_c = V_b$

$$\therefore \frac{P_c}{P_b} = \frac{T_c}{T_b} = \left(\frac{V_d}{V_b}\right)^n,$$

or if the ratio of expansion $\frac{V_d}{V_b}$ be called r, then $\frac{T_c}{T_b} = r^n$.

Hence,
$$E = 1 - n\frac{r - 1}{r^n - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

The work performed during the cycle is

$$L = J\,E\,Q_1 = J\left(1 - n\frac{r - 1}{r^n - 1}\right) Q_1. \quad . \quad . \quad (59)$$

If the suction-displacement corresponding to a heating-value Q_1 be designated D_q, then

$$D_q = V_d - V_b = V_b\left(\frac{V_d}{V_b} - 1\right) = V_b\,(r - 1). \quad . \quad . \quad (60)$$

The mean effective pressure becomes:

$$M.E.P. = \frac{J\,E\,Q_1}{D_q} = \frac{J\left(1 - n\frac{r - 1}{r^n - 1}\right)}{(r - 1)}\,\frac{Q_1}{V_b}. \quad . \quad (61)$$

By transformation of equation 57 there will be obtained

$$\frac{T_c}{T_b} = \frac{Q_1}{c_v\,T_b} + 1 = r^n. \quad . \quad . \quad . \quad . \quad . \quad (57a)$$

The maximum pressure will be

$$P_c = r^n\,P_b = P_b\left(\frac{Q_1}{c_v\,T_b} + 1\right). \quad . \quad . \quad . \quad . \quad (62)$$

II.

In the Otto cycle (see page 21), there is used as charge a mixture of a combustible gas and air of proper proportions, so that after compression and ignition the heating-value of the fuel becomes available for the heating of the working gases. When the specific heat of the gases of combustion is assumed to be the same as that of the fuel-gas and air (the weight of the gases before and after combustion being the same) the fact that the nature of

the working gases has changed through combustion does not influence the continuity of the cycle.

Figs. 20 and 21 represent a volume pressure and an entropy-temperature diagram of the cycle.

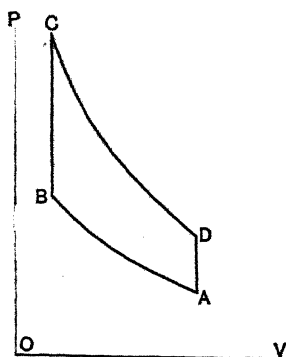



FIG. 20.

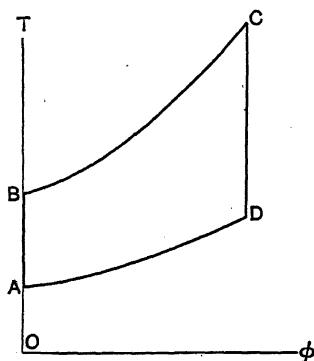



FIG. 21.

The events are:

From A to B. The charge is compressed, adiabatically, from the pressure of the atmosphere.

From B to C. Heat is transmitted; the volume remaining constant.

From C to D. Expansion takes place, adiabatically.

From D to A. The gas is discharged, or cooled at a constant volume.

The heat transmitted from B to C is

$$Q_1 = c_v (T_c - T_b), \quad . \quad . \quad . \qquad (63)$$

and the heat rejected from D to A is

$$Q_2 = c_v (T_d - T_a).$$

The efficiency of the cycle is

$$E = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_d - T_a}{T_c - T_b},$$

or

$$E = 1 \quad \frac{T_a}{T_b} \frac{\frac{T_d}{T_a} - 1}{\frac{T_c}{T_b} - 1}.$$

Due to the adiabatic compression and expansion we get, according to equation 33*a*,

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{n-1}$$

$$T_c = \left(\frac{V_a}{V_c}\right)^{n-1} T_d$$

$$T_b = \left(\frac{V_a}{V_b}\right)^{n-1} T_a.$$

But as $V_b = V_c$, and, in general, $V_a = V_d \therefore \frac{V_a}{V_b} = \frac{V_d}{V_c}$,

therefore $\frac{T_c}{T_b} = \frac{T_d}{T_a}$.

Hence, $E = 1 - \left(\frac{V_b}{V_a}\right)^{n-1} = 1 - \frac{1}{r^{n-1}}$; . . . . . . (64)

$\frac{V_a}{V_b}$ being designated by r.

The work performed during the cycle is

$$L = J.E.Q_1 = J\left(1 - \frac{1}{r^{n-1}}\right) Q_1. \quad . \quad . \quad . \quad (65)$$

If the suction displacement corresponding to the volume of one pound of the charge be D_q,

then, $$D_q = V_a - V_b = V_a\left(1 - \frac{1}{r}\right). \quad . \quad . \quad . \quad (66)$$

Hence $$M.E.P = \frac{J.E.Q_1}{D_q} = J \frac{1 - \frac{1}{r^{n-1}}}{1 - \frac{1}{r}} \frac{Q_1}{V_a} \quad . \quad . \quad (67)$$

The temperature range will be obtained by transposing equation 63, as

$$\frac{T_c}{T_b} = \frac{Q_1}{c_v T_b} + 1,$$

or $$\frac{T_c}{T_a} = r^{n-1}\left(\frac{Q_1}{c_v T_b} + 1\right). \quad . \quad . \quad . \quad (68)$$

The volume being constant during the period B to C we have

$$\frac{P_c}{P_b} = \frac{T_c}{T_b} = \frac{T_c}{T_a r^{n-1}},$$

Hence the maximum pressure

$$P_c = P_a r^n \left(\frac{Q_1}{c_v T_b} + 1\right). \quad . \quad . \quad . \quad . \quad (69)$$

III.

In the Brayton cycle (see page 24), the working charge is heated at a constant pressure until, at a certain point of the stroke regulated to suit the load, the heat-supply is cut off. From that point expansion is continued until the atmospheric pressure is reached, when, on the return stroke of the piston, the gases are

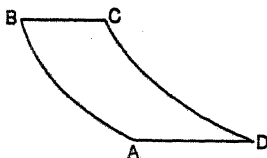
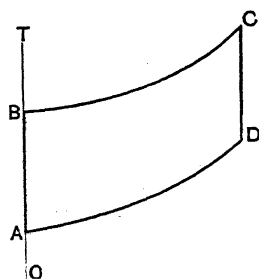



FIG. 22. FIG. 23.

discharged. Compression is effected from the pressure of the atmosphere to that of the initial working pressure in the cylinder.

Fig. 22 is a volume-pressure diagram and Fig. 23 an entropy-temperature diagram of the cycle.

The events are:

From A to B. The charge is compressed, adiabatically, from the atmospheric pressure.

From B to C. The heating of the charge is effected at a constant pressure.

From C to D. Expansion takes place, adiabatically, to the atmosphere.

From D to A. The gas is discharged, or cooled, at a constant pressure.

The heat transmitted during the period from B to C, at a constant pressure, is

$$Q_1 = c_p \ (T_c - T_b), \quad . \quad . \quad . \quad . \quad . \quad . \qquad (70)$$

and the heat rejected from D to A, also at a constant pressure, is

$$Q_2 = c_p \ (T_d - T_a).$$

The efficiency of the transformation during the cycle is:

$$E = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_d - T_a}{T_c - T_b},$$

or

$$E = 1 - \frac{T_a}{T_b} \frac{\frac{T_d}{T_a} - 1}{\frac{T_c}{T_b} - 1}.$$

The pressures being constant, the ratio of temperatures will be:

between B and C, $\frac{T_c}{T_b} = \frac{V_c}{V_b}$,

and between D and A, $\frac{T_d}{T_a} = \frac{V_d}{V_a}$.

For adiabatic compression and expansion we have, according to equation 33*a*,

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{n-1}$$

$$P_b = \left(\frac{V_a}{V_b}\right)^n P_a$$

$$P_c = \left(\frac{V_d}{V_c}\right)^n P_d.$$

But as $P_b = P_c$ and $P_a = P_d$

therefore $$\frac{V_a}{V_b} = \frac{V_d}{V_c}$$

and $$\frac{V_c}{V_b} = \frac{V_d}{V_a} = \frac{T_c}{T_b} = \frac{T_d}{T_a}.$$

Hence, $$E = 1 - \left(\frac{V_b}{V_a}\right)^{n-1} = 1 - \frac{1}{r^{n-1}}; \quad . \quad . \quad . \quad . \qquad (71)$$

when $\frac{V_a}{V_b}$ is designated by r.

If the suction-displacement corresponding to the volume of one pound of the charge be designated D_q,

then
$$D_q = V_d - V_b,$$

or
$$D_q = V_a\left(\frac{V_d}{V_a} - \frac{V_b}{V_a}\right).$$

But as
$$\frac{V_d}{V_a} = \frac{V_c}{V_b} = \frac{T_c}{T_b} = \frac{T_d}{T_a},$$

therefore
$$D_q = V_a\left(\frac{V_c}{V_b} - \frac{V_b}{V_a}\right) = V_a\left(\frac{V_c}{V_b} - \frac{1}{r}\right). \quad . \quad . \quad . \quad (72)$$

The work performed during the cycle is

$$L = J.E.Q_1 = J\left(1 - \frac{1}{r^{n-1}}\right)Q_1 \quad . \quad . \quad . \quad (73)$$

Hence
$$M.E.P. = \frac{J\,E\,Q_1}{D_q} = \frac{J\left(1 - \frac{1}{r^{n-1}}\right)Q_1}{\left(\frac{V_c}{V_b} - \frac{1}{r}\right)V_a} \quad . \quad . \quad (74)$$

The ratio $\frac{V_c}{V_b}$ being necessarily larger than the unit, the M.E.P. for the Brayton cycle is lower than that for the Otto cycle.

The temperature range will be obtained by transposing equation 70.

$$\therefore \frac{T_c}{T_b} = \frac{Q_1}{c_p T_b} + 1,$$

or
$$\frac{T_c}{T_a} = r^{n-1}\left(\frac{Q_1}{c_p T_b} + 1\right). \quad . \quad . \quad . \quad . \quad . \quad (75)$$

The maximum pressure is

$$P_c = r^n P_a. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

IV.

Carnot's cycle, as applied to a perfect gas, is shown in the volume-pressure diagram Fig. 24 and in the entropy-temperature diagram Fig. 25.

The events are:

From A to B. Adiabatic compression from the atmospheric pressure.

From B to C. Heat is added isothermally.
From C to D. Adiabatic expansion to a pressure below the atmospheric pressure.
From D to A. Cooling is effected, isothermally, to the original volume and pressure.

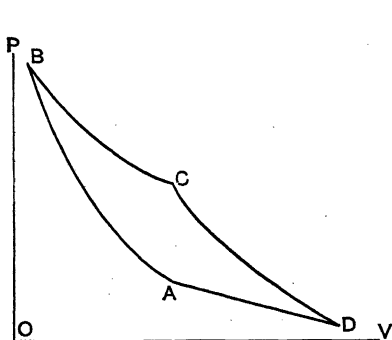
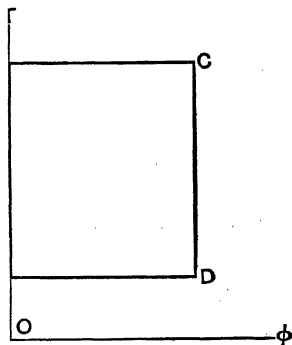



FIG. 24. FIG. 25.

According to equation 5, page 6, we have

$$E = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{T_d}{T_b}.$$

But $T_d = T_a$ and, due to the adiabatic compression, we have

$$T_b = \left(\frac{V_a}{V_h}\right)^{n-1} T_a = r^{n-1} T_a; \quad \frac{V_a}{V_h} \text{ being designated } r.$$

Hence, $$E = 1 - \quad . \quad . \quad . \quad . \quad . \quad . \quad (77)$$

The heat rejected during the isothermal compression from D to A is, according to equation 30,

$$Q_2 = (c_p - c_v)\, T_d \log_e \frac{V_d}{V_a},$$

and as $$Q_1 = \frac{T_b}{T_d} Q_2,$$

therefore $$Q_1 = (c_p - c_v)\, T_b \log_e \frac{V_d}{V_a}$$

$$\therefore \frac{V_d}{V_a} = e \quad ,$$

e being the base for the hyperbolic logarithms = 2.718.

The suction displacement corresponding to a volume of one pound of charge being designated D_q, we have

$$D_q = V_d - V_b = V_a \left(\frac{V_d}{V_a} - \frac{V_b}{V_a}\right),$$

or

$$D_q = V_a \left(e^{\frac{Q_1}{(c_p - c_v)\, T_b}} - \frac{1}{r} \right). \quad . \quad . \qquad (78)$$

The work obtained per pound mixture is

$$L = J E Q_1 = J \left(1 - \frac{1}{r^{n-1}}\right) Q_1. \quad . \quad . \qquad (79)$$

Hence,

$$M.E.P. = \frac{J E Q_1}{D_q} = J \cdot \frac{\left(1 - \frac{1}{r^{n-1}}\right) \quad Q_1}{\frac{Q_1}{(c_p - c_v)\, T_b} \quad 1} \qquad (80)$$

The temperature range is

$$\frac{T_c}{T_a} = r^n \qquad (81)$$

and the maximum pressure

$$P_b = r^n P_a. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (82)$$

By the results obtained it will be evident that Cycle I is unfavorable compared with the others, due to low efficiency and low mean pressure, and it is, at present, therefore, never used.

Cycles II, III, and IV are theoretically of the same efficiency, and would be equally serviceable in respect to returns given, but for the fact that in practice the cycle of the highest temperature range would dissipate more heat through transmission and radiation. A low temperature range is, therefore, favorable. The size of the engine will depend on the mean effective pressure of its cycle, and in this respect the Otto cycle is the most favorable.

The Cycles II, III, and IV compare as follows:

Efficiency.	Low.	Intermediate.	High.
Temperature range.....	Carnot	Brayton	Otto
Pressure range.........	Brayton	Carnot	Otto
M. E. P.	Carnot	Brayton	Otto

The principal cycles involve, as has been seen, each, three or four distinct events represented in the diagrams by lines, each of which follows its own defined law. These lines have in the preceding been referred to as the adiabatic, the isothermal, the constant-pressure, and the constant-volume line. The latter two lines are often referred to by different names. Referring to Figs. 18 and 19 the line $D\,B$ of the $\phi\,T$-diagram is a constant-pressure line, isopiestic line or isobar, and it will of course always in the $V\,P$-diagram be a horizontal line, as $B\,A$, Fig. 18. The line $B\,C$, in the $\phi\,T$-diagram, is a constant-volume line or isometric line, and it becomes, in the $V\,P$-diagram, always a vertical line.

Many cycles have been tried, or proposed, which involve a greater number of events than the four considered in the previous, but, being complicated, they can, at present, be considered only as experiments. For a special study of different cycles reference should be made to "The Gas Engine," by F. R. Hutton.

CHAPTER IV

POWER, SIZE, AND SPEED OF GAS-ENGINES

Horse-Power, Overload Capacity.—When determining the required size of a gas-engine cylinder for a specified power and fuel, it is generally the case that previous experience with the fuel in question has decided what mean effective pressure may safely be figured on. Should any data in this respect not be at hand, then it will be necessary to determine the expected mean pressure according to the method described at page 119.

We assume, for the present, the mean effective pressure of a normal representative indicator card as given, and this pressure, expressed in pounds per square inch, we shall in the following denote by the symbol P_{mc}, signifying that it is the mean effective pressure as given by a normal indicator card.

In a single two-cycle engine one explosion occurs every second stroke, and in a single four-cycle engine one explosion occurs every fourth stroke. An equivalent mean effective pressure, assumed to act on the piston continuously throughout the whole cycle, would be:

in a single two-cycle engine $\frac{1}{2} P_{mc}$, and

in a single four-cycle engine $\frac{1}{4} P_{mc}$.

In a multiple-cylinder single-acting or double-acting engine there may occur one or more explosions at every, or every second, stroke, depending on the type of the engine; wherefore, written in a general form, the total mean effective pressure acting continuously for every revolution, will be:

for a two-cycle engine $\frac{\epsilon}{2} P_{mc}$, and

for a four-cycle engine $\frac{\epsilon}{4} P_{mc}$;

ϵ signifying the number of explosion-chambers the engine includes.

The indicated horse-power for the two types of engines will be:

for a two-cycle engine

$$I.H.P. = \epsilon \frac{P_{mc} A S}{2 \times 33,000} = \epsilon \frac{P_{mc} L A N}{33,000} \quad . \quad . \quad (83)$$

for a four-cycle engine

$$I.H.P. = \epsilon \frac{P_{mc} A S}{4 \times 33,000} = \epsilon \frac{P_{mc} L A N}{33,000 \times 2} \quad . \quad . \quad (84)$$

S being the piston-speed in feet per minute,
L the length of the stroke in feet,
A the area of the piston in square inches,
N the number of revolutions per minute, or
the number of explosions per minute, in a two-cycle engine;
$\frac{N}{2}$ the number of explosions per minute, in a four-cycle engine,
and ϵ the number of explosion-chambers the engine includes.

The mean effective pressure that can be obtained in a gas-engine, the factor P_{mc}, is a somewhat uncertain quantity, varying to some extent in cards taken successively on a constant load. If the P_{mc} inserted in the formula is the representative pressure that experience shows will be obtained under favorable circumstances at full load, then the corresponding I.H.P. may be said to be the maximum capacity of the engine.

It is required, however, that an engine shall have some over-load capacity to allow for occasional short heavy impulses of the resistance, or to allow for occasionally occurring poor quality of gas-mixture. It is, therefore, customary to rate the engine somewhat below its maximum capacity, and normally an allowance of 15 per cent overload capacity would be considered proper. With this allowance, a hit-or-miss engine, running at rated load, would cut out the charge fifteen times of every one hundred explosion strokes, or it would miss one in about every eight explosion strokes.

The allowance for overload capacity can of course be made in the mean effective pressure inserted in the formula, by assigning to it a value about 15 per cent less than what experience shows can be safely obtained, or a coefficient C, to be assigned a suitable

value, may be inserted in the formula for the rated indicated horse-power.

We obtain, thus, for a four-cycle single-cylinder single-acting engine

$$\text{rated I.H.P.} = C \frac{P_{mc} L A N}{33{,}000 \times 2}. \quad . \quad . \quad . \quad (85)$$

The coefficient C, corresponding to different percentages overload capacity, values as follows:

Overload capacity	10 %,	15 %,	20 %,	25 %.
C	0.91	0.87	0.83	0.8.

The internal resistance in an engine, or the friction of piston, stuffing-boxes, journals, etc., and resistance of valves and springs, will cause a loss in power of 10 to 20 per cent of the indicated horse-power. The brake horse-power, or the power that can be taken off from the engine-shaft by means of a brake and brake-wheel, will, accordingly, be 10 to 20 per cent less than the indicated horse-power, or as much less as the loss due to internal resistance. The ratio between brake horse-power and indicated horse-power is the mechanical efficiency, and if the latter be denoted by m we have

$$\text{Mechanical efficiency} = m = \frac{B.H.P.}{I.H.P.}.$$

The efficiency, m, varies between 0.80 and 0.90, but in an average case it would be safe to assign to it a value 0.85.

The maximum brake horse-power of a four-cycle single-cylinder single-acting engine becomes

$$\text{max. B.H.P.} = m \frac{P_{mc} L A N}{33{,}000 \times 2}, \quad . \quad . \quad . \quad (86)$$

and its rated brake horse-power

$$\text{rated B.H.P.} = m.C \frac{P_{mc} L A N}{33{,}000 \times 2}. \quad . \quad . \quad . \quad (87)$$

For an engine of 15-per-cent overload capacity and whose mechanical efficiency is 0.85 we get the coefficient $m\,C = 0.85 \times 0.87 = 0.74$.

It will be recognized that these power-formulas are identical with those for the steam-engine, excepting that the number of

impulses per minute, $2N$, in the steam-engine is replaced by the number of explosions, $\frac{N}{2}$, in the gas-engine.

For the sake of simplicity, the preceding power-formulas, excepting the first two, refer to single-cylinder single-acting four-cycle engines, or refer to one explosion-chamber only. It is evident, that if, instead of one single-acting cylinder, or one combustion-chamber, we employ two single-acting, or one double-acting cylinder, there would be obtained twice the power, and hence, when two or more combustion-chambers are employed in a four-cycle engine, the total result will be 2, or ϵ times that expressed by the formulas.

In a two-cycle engine the power becomes always double that of a four-cycle engine of the same type.

Piston Speed.—It is often found, particularly when an even fuel-mixture is hard to control on a variable load, that an engine is liable to back-fire at occasions of light loads. This is due, generally, to slow combustion of the charge; the cylinder still containing hot combustion-products and fire when the inlet valve opens. A high piston-speed is in such cases undesirable for two reasons. Partly because, on the suction-stroke, the mixing of the fuel and air (and vaporization of the fuel in case it is liquid) becomes less complete, and partly because there will be less time for the exhausting and cooling down of the burned gases.

A high piston-speed is also unfavorable on slow-burning mixtures, because, when the flame-propagation is slow, the maximum pressure may not be reached before the piston is well toward the middle of its stroke, and hence, the resulting efficiency becomes low. Further, the piston and exhaust valve, when not cooled, become, at high piston-speed and at a great number of revolutions per minute, considerably heated and apt to cause self-ignition of the charge. A high piston-speed may be said also to be objectionable on account of the increased wear of the cylinder, piston, and rings that it will cause.

On the other hand, a better economy will be realized, under normal conditions, when the piston-speed is high, due to the fact

that a smaller percentage of heat will become dissipated through the cylinder walls into the jacket water. And further, the higher the speed the greater the effect of an engine of a given size will be; hence, a smaller and lighter engine will do the same work at a high speed as a larger and heavier one at a slower speed. The economizing in weight and size becomes, of course, in certain classes of engines, of the greatest importance.

When pre-ignitions are caused by a too high compression of the fuel, it may be possible that an increased speed of a short-stroke engine will overcome the trouble. The reason is, that as the speed is increased the ignition may properly be advanced to an earlier point of the cycle, hence the point at which the self-ignitions occur may finally be suitable for the pace to which the engine is speeded. The ignition in this case is generally caused by some unjacketed part in the combustion-space, probably the exhaust valve or piston, which remains hot enough to cause regular ignitions. As a proof, it is sometimes found that a high-speed engine, once well started, may continue to run quite satisfactorily even if the regular ignition-device be thrown out of action.

The limitations of the piston-speed, with regard to all of these considerations, must be made with guidance from past experience with different engines, and in the following table there will be found some average speeds that have proven safe for various types of motors.

TABLE III

Usual Piston Speeds for Various Classes of Gas-Engines

Heavy stationary engines, 30 H.P. or over, with separate mixing-chambers for each cylinder .	700– 950 ft. per min.
Small stationary engines with separate mixing-chambers for each cylinder	400– 600 " " "
Multiple-cylinder engines with common mixing-chamber	500– 700 " " "
Two-cylinder automobile engines	600– 800 " " "
Multiple-cylinder automobile engines	800–1,000 " " "

Piston-Speed with Reference to Compression and Weight of the Reciprocating Parts.—To exclude undesirable knocks and

undue wear in the connecting-rod journals, the piston speed must be limited to suit the weight of the reciprocating parts and the compression carried.

This requirement should be carefully considered in connection with the development of a new line of engines.

The pressure per square inch of piston-area required for opposing the maximum inertia-effect of the reciprocating parts, at the end of each stroke, is, according to formulas 101*h* and 101*c*, page 195:

At the head-end of the cylinder $\frac{P_1}{F} = 0.000034 \frac{G}{F} N^2 r,$

and at the crank-end of the cylinder $\frac{P_2}{F} = 0.000023 \frac{G}{F} N^2 r.$

Assume that in a given case the stroke of the piston is 24 inches, the weight of the reciprocating parts per square inch piston area is 3 pounds, the number of revolutions per minute 300 and the desired compression of the charge 70 pounds.

The maximum accelerating force, per square inch piston-area, will be:

at the head-end, $\frac{P_1}{F} = 0.000034 \times 3 \times 90{,}000 \times 12 = 110$ pounds,

at the crank-end, $\frac{P_2}{F} = 0.000023 \times 3 \times 90{,}000 \times 12 = 76$ pounds.

Offset these pressures on the perpendiculars $O P_1$ and $A P_2$; respectively, on the negative and positive side of the base-line $O A$, Fig. 26. Locate the point x,* $0.08r$ from the middle ordinate $n n$, and draw the inertia curve $P_1 x P_2$ through the points P_1, x, and P_2.

Draw, further, the compression curve $A B$ for a compression from the atmospheric pressure to a pressure of 70 pounds gauge, and combine the compression curve and the inertia curve into a line $E o P_2$. It will then be noticed that the pressure on the back of the piston becomes negative at the point o. The piston and the connecting-rod will, therefore, at that point tend to fly out away from crank, and while the contact-surface between the

* According to page 197.

crank-pin and crank-pin box has during the period from A to o been on the side of the pin toward the cylinder, it will at the point o change over to the opposite side. At point o there will, under these conditions, due to back-lash, be a decided knock in the crank-pin and in the cross-head-pin. These undesirable sudden changes of contact-surface in the journals, during periods when they are exposed to the working strains, can be avoided by arranging conditions so that there will always, during the com-

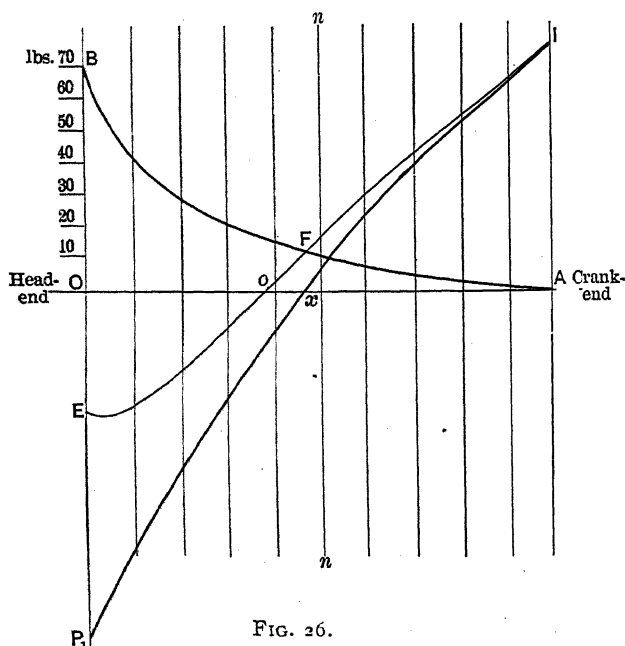



FIG. 26.

pression-stroke, be a surplus pressure back of the piston, above that required to resist the inertia-force, to press the piston and connecting-rod boxes against the back of their respective pins. If this is done then the combined line of pressure on the crank-pin will never cut down on the negative side of the baseline $O\,A$.

Fig. 27 is a combined pressure and inertia diagram in which the positive pressure on the piston, due to the compression, just balances the negative force, due to the inertia of the reciprocating parts, at the point P.

The compression curve is the same as in Fig. 26, and the speed for which the inertia curve is drawn is 210 revolutions per minute. The point P of the lowest combined pressure on the crank-pin, due to compression and inertia, occurs at an angular distance of about 40 degrees from the head-end centre, and the pressure, P_1, required at the head-end of the stroke for opposing the inertia is about ¾ of the compression pressure.

The initial acceleration pressure P_1 required to bring the combined curve down to the baseline OA, at P, becomes for

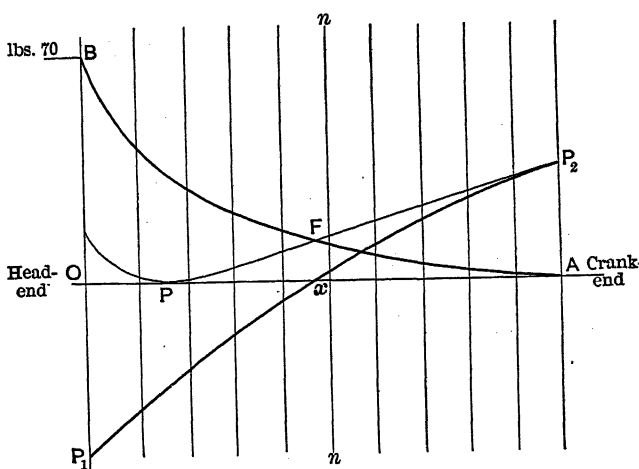



FIG. 27.

high compression ratios somewhat less than ¾ of the compression pressure, but, as it is low-compression high-speed engines, particularly, that need to be investigated with reference to the limitation of speed to suit the compression, we may employ in the formulas the fraction ¾.

Thus, if C is the compression pressure (70 to 120 pounds per square inch), then it is required that

$$\frac{P_1}{F} < \frac{3}{4} C,$$

or

$$0.000034 \frac{G}{F} N^2 r < \frac{3}{4} C.$$

If the mean piston speed, in feet per minute, be called S, and because r is expressed in inches, we have

$$\frac{4 r N}{12} = S,$$

or

$$r N = 3 S,$$

which inserted in the leading expression gives

$$3 \times 0.000034 \frac{G}{F} N S < \frac{3}{4} C, \text{ or}$$

$$S < \frac{7{,}353\, C}{\frac{G}{F} N} \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

S = maximum piston speed allowable, in feet per minute.
C = the compression pressure, in pounds per square inch.
$\frac{G}{F}$ = weight of reciprocating parts, per square inch in piston-area (including piston, piston-pin, and connecting-rod).
N = number of revolutions per minute.

The actual piston speed adopted must be less than S by a proper margin.

By means of the preceding formula it can readily be ascertained that in order to run, for instance, an automobile engine smoothly at a very high piston speed, say, at 1,000 feet per minute, it will be very necessary to make the reciprocating parts as light as possible.

Let, for instance, it be required to determine the proper limit for the weight of the piston and connecting-rod for an engine to run at a piston speed of 1,000 feet per minute; the length of the stroke being 9 inches and the compression 80 pounds.

$$\text{We have } \frac{G}{F} < \frac{7{,}353\, C}{S N}$$

$$\text{and } \frac{3 S}{r} = 666 \text{ revolutions per minute.}$$

$$\text{Hence, } \frac{G}{F} < \frac{7{,}353 \times 80}{1{,}000 \times 666},$$

or $\frac{G}{F} < 0.9$ pounds per square inch piston-area. $\frac{G}{F}$ being re-

quired to be less than 0.9 pounds per square inch piston-area, it would probably be made $\frac{8}{10}$ pound per square inch, or, if possible, even less.

The allowable maximum weight of the reciprocating parts increases, it will be noticed, with the length of the stroke, so that for an engine of a 32-inch stroke, for instance, and of the same piston speed, 1,000 feet per minute, the maximum weight of the reciprocating parts would be 3.2 pounds per square inch piston-area.

The allowable weight increases also in proportion to the compression pressure, wherefore the same 32-inch-stroke engine, if working with a compression of 120 pounds per square inch, could be allowed a maximum weight of the reciprocating parts of 4.8 pounds per square inch of piston-area.

For tandem engines, particularly when of large proportions and provided with water-cooled pistons, it will not be possible, with reasonable high piston speed, to make the reciprocating parts light enough to comply with the conditions for smooth running imposed by the above formula.

The piston-speed and number of revolutions for such engines should, however, be chosen as moderate as circumstances will allow; an average should be drawn between the advantage that may be gained from a higher efficiency and reduced initial cost and the disadvantage resulting from a less smooth action during the compression-strokes.

Efficiency and Compression.—The equation for the efficiency of the theoretical gas-engine cycle is

$$E = 1 - \left(\frac{V_b}{V_a}\right)^{n-1} \text{ or}$$

$$E = 1 - \frac{1}{r^{n-1}};$$

V_a being the total volume of the cylinder, V_b the volume of the compression space, and r the ratio $\frac{V_a}{V_b}$.

It appears, thus, that the theoretical efficiency increases with the intensity with which the charge is compressed before it is ignited. On the other hand, the nature of the fuel-gas used may

limit the compression that can be employed, so that the maximum economy obtainable will depend, finally, on the maximum compression that the fuel can stand without becoming liable to cause pre-ignitions.

The thermal efficiency of the practical heat-transformation, expressed by the coefficient Efy, is, thus, in the first place dependent on the compression that can be allowed, but it is also affected by the amount of heat that will be conducted away from the hot gases through the walls of the cylinder in which they are working; wherefore, a high compression, and consequently high initial temperature, will again, due to an increased heat-loss to the jacket water, tend to reduce the thermal efficiency.

Again, some fuels form a proper explosive mixture which contains much more heating-value per unit volume than others, and they, requiring less cylinder-volume and therefore less cooling-surface per horse-power, tend, other things being equal, to show a higher thermal efficiency.

The proportioning of the valve-port areas and the timing of the opening and closing of the valves, which have influence on the amount of negative work done during the cycle, will also affect to some extent the thermal efficiency of an engine.

From what has been stated it is evident that the question regarding the efficiency obtainable in a gas-engine is a complicated one, and it can hardly be solved, finally, excepting through experiments with each particular fuel.

It is quite possible, however, to foretell in a general way the degree of economy that should be realized, when the compression pressure that can be employed is definitely known.

Under normal good conditions, with a compression carried to 25 per cent of the initial volume, an efficiency of 0.24 should be expected. With an increased compression, under the same conditions, the efficiency would increase in a rate approximately as per the following Table IV.

One horse-power being 33,000 foot-pounds per minute and 778 foot-pounds being equivalent to one heat-unit, there will be required 2,545 heat-units per hour for obtaining one indicated horse-power, assuming that no heat-loss is sustained during the

transformation. If the thermal efficiency of the transformation be 24 per cent, an expenditure of 10,600 B.T.U. per indicated horse-power would be required per hour. The economy of the engine would in this case be said to be 10,600 B.T.U. (The low heating-value of the fuel should be counted.) To each efficiency coefficient, $E f y$, there corresponds, thus, a certain economy expressed in heat-units per hour, obtained from the relation

$$\text{economy} = \frac{2{,}545}{E f y}.$$

In Table IV are given the values for the economy corresponding to various compression ratios and efficiencies.

The ratio between the work represented by the theoretical cycle and the work represented by the indicator-card is

$$f y = \frac{E f y}{E}.$$

The heat-units consumed per indicated horse-power per hour, which is the economy of an engine, being known, and the factor E being determined from the compression-ratio, the factor $f y$ may be obtained from the expression

$$f y = \frac{2{,}545}{E \times \text{H. U. cons'd per I. H. P. per hour}}.$$

This factor, in the steam-engine practice referred to as the diagram-factor, is with respect to the gas-engine much smaller than what it is with respect to the steam-engine. With respect to the gas-engine it has been found to vary between the values 0.4 to 0.64; the latter figure to be considered very favorable.

For comparison between results obtained by different gas-engines the diagram-factor is more suitable as a basis of reference than the thermal-efficiency coefficient, as the advantage one engine may have over another, on account of a higher compression, is eliminated.

In Table IV are also given the values of the factor $f y$, which correspond to the factors E and $E f y$ quoted.

These figures do not include, however, the mechanical efficiency of the motor, but only its thermal efficiency at trans-

forming heat into power exerted on the piston, or into indicated horse-power. To obtain the expected efficiency at transforming heat into brake horse-power the figures of the table should be multiplied by the factor for the mechanical efficiency, which may be assumed, on an average, 0.85.

TABLE IV.

Normal Efficiency and Economy for Varying Compressions.

Compression ratio r	3.5	4	4.5	5	5.5	6	6.5	7	7.5
Compres'ion gauge	62	76	90	105	121	138	155	172	190
Theoretical effi'cy E	0.32	0.354	0.380	0.404	0.426	0.447	0.466	0.482	0.496	0.507
Thermal effi'cy, E/y	0.19	0.22	0.24	0.255	0.268	0.28	0.29	0.30	0.305	0.31
Factor $/y$	0.6	0.62	0.63	0.63	0.63	0.63	0.62	0.62	0.62	0.61
Economy	13400	11600	10600	10000	9500	9100	8750	8400	8300	8200

The Compression Curve.—During the compression of a gas-mixture, from its initial pressure at the end of the suction-stroke, to the pressure at which it is desired to explode the charge, the pressure in the cylinder increases approximately in accordance with the adiabatic compression-line, the formula for which is:

$$\frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n$$

p_1 being the initial pressure,

v_1 the total cylinder volume,

p_2 the pressure at the various points of the curve,

v_2 the volumes corresponding to the pressures p_2,

and n the exponent for the compression curve.

The exponent n is the ratio between the specific heat of the gas at constant pressure and that at constant volume, and for a perfect gas it is very close to 1.4.

Due to the fact that the gas, during the latter part of the compression in the gas-engine cylinder, transmits some of its heat to the water-jacket, its compression curve will fall somewhat lower than that for a perfect adiabatic compression, and the exponent n will average the value 1.35.

Determination of the Value of the Exponent n.—The exponent

n for a given gas-mixture is readily determined in practice, from carefully made observations of the pressure obtained in a measured clearance space.

For instance:—In a 14¾ × 24 engine running on producer-gas the compression pressure was 160 pounds, measured by a gauge and by the indicator. The volume of the clearance space, determined by measuring the volume of water required to fill it, was 678 cubic inches.

The piston displacement amounts to 24 × 170.87 = 4,101 cubic inches.

The volume of the clearance space = 678 cubic inches,

The total cylinder-volume, therefore, = 4,779 cubic inches.

The pressure in the cylinder, at the end of the suction-stroke, was 1.7 pound vacuum, or 13 pounds absolute.

The equation for adiabatic compression is:

$$\frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n$$

Hence,

$$\frac{13}{174.7} = \left(\frac{678}{4{,}779}\right)^n$$

$$\log 13 - \log 174.7 = n\,(\log 678 - \log 4{,}779)$$

$$\begin{array}{r} 1.11394 \\ -2.24229 \\ \hline 0.87165-2 \end{array} = n \left\{ \begin{array}{r} 2.83123 \\ -3.67934 \\ \hline 0.15189-1 \end{array} \right.$$

$$n = \frac{1.12835}{0.87681} = 1.33$$

In Fig. 28 are drawn two sets of compression curves. One set starting from the atmospheric pressure, 14.7 pounds, and one starting from an absolute pressure of 13 pounds. The indexes for each set of curves are, respectively, $n = 1.33$ and $n = 1.37$.

The pressure at the beginning of the compression stroke, at full speed of the engine, being always somewhat below the atmospheric pressure (often one to two pounds), the final compression pressure will, therefore, ordinarily fall between the two lower curves when the engine is up to speed. At a slow speed, however, the compression pressure will more likely fall between the two upper lines.

ABSOLUTE PRESSURES
200 195 190 185 180 175 170 165 160 155 150 145 140 135 130 125 120 115 110 105 100 95 90 85 80 75 70 65 60 55 50 45 40 35 30 25 20 15 10 5 0
VOLUMES
10% 20% 30% 40% 50% 60% 70% 80% 90% 100%
n=1.37
n=1.33
n=1.33
n=1.37
COMPRESSION-LINES
FOR n=1.33 AND FOR n=1.37
EACH SET OF LINES BEGINS:
AT THE PRESSURE 14.7 POUNDS
AND AT THE PRESSURE 13.2
POUNDS

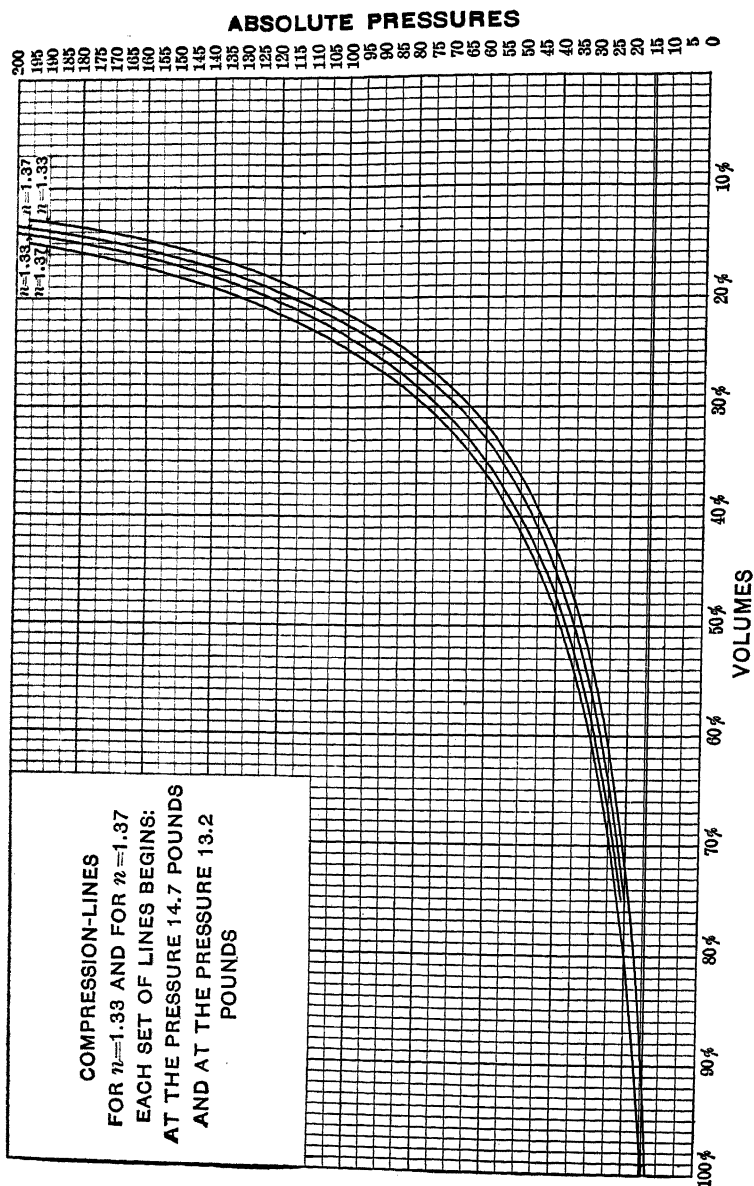



FIG. 28.

It will be noticed that, at high compressions, there is a considerable difference between the pressure obtained when the compression begins at the atmospheric pressure, and that obtained when it begins at a pressure somewhat below the atmosphere. It is, therefore, at high compressions, impossible to foretell exactly what the compression pressure actually will be, and it may even vary quite materially for different strokes of the engine. However, it can, by means of the diagram, always be determined closely enough for practical requirements.

A correct value of the exponent n cannot be obtained from indicator cards, excepting when the clearance space of the cylinder is very carefully determined, and when the valves are known to be tight. To assign to the exponent n a likely value, and figure the clearance space from the expansion line, leads generally to preposterous results.

The cards, Fig. 29 and 30, are from engines working on producer gas, the clearance spaces of which were determined carefully. When figuring the exponent n from the ordinates, which are drawn on the cards, the following results were obtained:

Exponent n for card Fig. 29.

figured between ordinates a and b $n = 1.33$.
figured between ordinates a and c $n = 1.31$.
figured between ordinates a and d $n = 1.30$.
figured between ordinates a and e $n = 1.27$.
figured between ordinates d and e $n = 1.23$.

This card shows heat to be gained during the expansion.

Exponent n for card Fig. 30.

figured between ordinates a and b $n = 1.29$.
figured between ordinates a and c $n = 1.32$.
figured between ordinates a and d $n = 1.33$.
figured between ordinates a and e $n = 1.35$.
figured between ordinates d and e $n = 1.45$.

This card shows a loss of heat during the expansion.

Such variation in the value of n as shown by these figures is

not any exception, but it is found to be more or less the rule for all cards. The figured values of n vary often in practice from as low as $n = 1.15$ to a value $n = 1.45$, but a good part of the variation may be due to the difficulty of measuring the ordinates

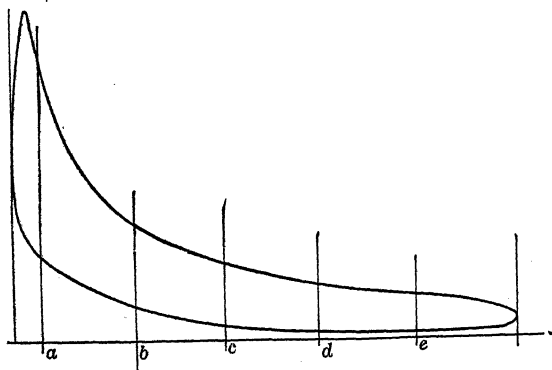



FIG. 29.

of the card correctly. In some instances the discrepancy is due to leaky valves, and in such cases the compression line is also affected.

Figs. 31 and 32 are cards taken with the igniter wire dis-

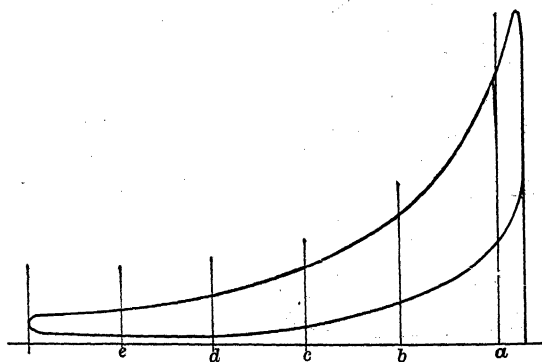



FIG. 30.

connected, no explosion therefore taking place. The charge is merely compressed, according to the line $a-b$, and expanded again according to the line $b-c$. The card, Fig. 31, shows the valves to be tight, whereas the card, Fig. 32, shows leaky valves.

Due to the cooling of the charge, there is always some work lost in the compression, which is represented by the area enclosed between the compression and expansion lines.

When taking a series of compression cards it is often the case that some cards show considerably greater tightness of the valves than others, which cannot very well be explained on other grounds than that the valves seat themselves less tight during occasional

FIG. 31.

strokes. This will, of course, have effect also on the regular power cards during these strokes.

Compression cards may readily be obtained, as stated, by disconnecting the igniter after the engine is up to speed, or in a multiple-cylinder engine one cylinder at a time may be tested for compression, while the other cylinders keep the engine up to speed. There would, in a multiple-cylinder engine, be some

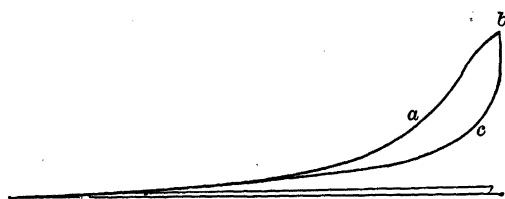



FIG. 32.

risk of exploding the unconsumed charge in the exhaust-pipe, when the igniter for one cylinder is thrown out of action. To prevent this, the gas may be closed off and only a full charge of air taken in for compression. However, if the air becomes wire-drawn to a greater extent than the regular charge this will affect the density of the material actually being compressed and will give a result different from that of the regular charge.

The compression pressure that can safely be used for different fuels depends essentially on the type and construction of the engine, and only experience with different engine-types and different fuels can finally decide upon the most favorable compression to use.

In the following table are recorded compression pressures, for various fuels, that are commonly used in modern practice:

	Pounds, gauge
Natural gas	100 to 150
Coal gas or manufactured city gas	80 to 140
Coke-oven gas	100 to 140
Bituminous producer gas	140 to 160
Anthracite producer gas	150 to 170
Blast-furnace gas	150 to 190
Carbureted gasoline	60 to 90
Kerosene	50 to 80
Alcohol	120 to 180

The higher of these pressures are somewhat higher than those quoted by Prof. C. E. Lucke,* but the tendency has been to raise the compression, wherever, through suitable regulation of the mixture, and through cooling of exhaust valves and pistons, this could be done.

Producer-gas engines built especially for this fuel seldom use less than 160 pounds compression, but in the latest large blast-furnace gas-engines the compression has been cut down to 150 pounds.

Scavenging.—In the two-cycle engine, the burned gases from a preceding stroke are generally removed from the combustion-chamber by the new charge, which is admitted under a slight pressure. Occasionally a compressed charge of pure air is used in connection with four-cycle engines for the purpose of removing the neutrals (to effect scavenging). The object of scavenging would be either to increase the capacity of the cylinder by making it possible to admit a denser charge consisting of pure air and fuel gas only, or, in special cases, to obviate pre-ignitions by admitting an initially cooler charge.

Admitting that a forced scavenging of the combustion-chamber

* C. E. Lucke, "Gas-Engine Design."

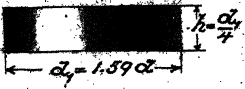
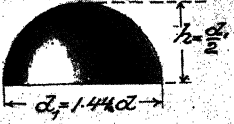
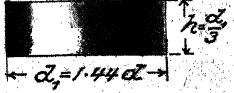
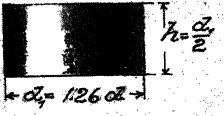
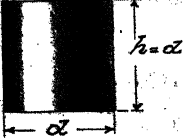
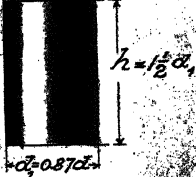
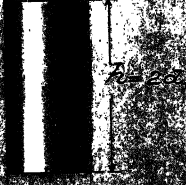
will permit a higher compression pressure to be used, it would tend toward increasing the efficiency. However, practice does not always show any decided gain in this respect, probably on account of greater losses due to increased fluid velocities, or due to friction-losses in the necessary compression apparatus. Forced scavenging is, therefore, not generally employed, excepting in two-cycle engines.

It is possible to effect, to some extent, automatic scavenging, simply by a close adjustment of the inlet- and exhaust-valves to suit the velocity and inertia-effect of the exhaust gases. It is evident that the inertia of the column of exhaust gases, their velocity being considerable although their weight is not great, must have a tendency to cause a slight vacuum in front of the piston at the end of its return exhaust stroke. This tendency may be taken advantage of by judiciously opening the inlet valve slightly before the crank passes the dead centre, and by not closing the exhaust until the centre is well passed, thus giving the body of exhaust gases in the discharge-pipe a chance to spend its inertia in drawing out the neutrals and in pulling in the new charge. Inside proper limits, the longer and the straighter the discharge pipe is the greater its effect will be on the scavenging.

The Length of the Stroke.—The proper ratio between the length of travel and the diameter of the gas-engine piston is often stated, arbitrarily, to be between 1.2 and 1.6. The choice of the ratio between these limits is, it is true, of no great importance, and the disposition of the subject in a general statement is satisfactory, when it concerns only engines of uniform and low compression. The subject deserves, however, to be considered with more care when embracing modern engines which, for some fuels, employ a very high compression.

In connection with the subject of compression it was brought out that, in order to secure best economy, the object should be to use as high compression as the fuel will allow without becoming liable to cause pre-ignitions, and it may be said that suitable compression pressures for different fuels vary, on an average, between 45 and 200 pounds gauge per square inch.

Assuming that the same engine would be required to work,

at alternate occasions, with either one of these compression pressures, then a clearance space of 50 per cent of the displacement would be required for giving the lower pressure, whereas for the higher pressure a clearance of only 13 per cent would be proper. For example: in a 12 × 16 cylinder the clearance in the first instance would be a volume 12 inches in diameter by 8 inches high, and in the second instance it would be a volume 12 inches in diameter by 2 inches high. In other words, the length of a cylindrical clearance-volume would in the first instance be ⅔ of the diameter, and in the second ⅙ of the diameter. The inconsistency of these two ratios between the diameter and length of the clearance-volumes becomes evident from the fact that the cooling-surface enclosing the smaller combustion space is, per unit volume, about 130 per cent larger than the cooling-surface enclosing the larger one.

If it be granted that an increased economy will result from having the combustion-chamber enclosed by the smallest possible cooling-surface, then it is evident that the ratio, 12 inches diameter of the cylinder by 16 inches stroke, is not a favorable one when a high compression pressure is used, and that it will be improved by increasing the stroke.

In Table V are represented various bodies, cylindrical in form excepting Fig. 2, of the same cubical contents, but of different ratios between diameter and height. It will be noticed that the enclosing surface is the smallest when the ratio is as 1 to 1, and that it increases as the body is elongated or depressed.

The working gases in a cylinder being of the highest temperature at the time of completed combustion, it is more important to have them enclosed by the least cooling-surface per unit volume during the combustion and early part of the expansion stroke, rather than during the latter part of the stroke. Hence the shortest height of the combustion space should be limited to that when a ratio of about 3 to 1 is reached; the cooling-surface, per unit volume, from there, increasing appreciably.

A half sphere of the same volume, as the cylindrical volume Fig. 2 of the table, has a surface 1.555 against 1.72 of the cylinder. A spherically formed combustion space would therefore properly be

TABLE V.

Figure.	Volume.	Enclosing Surface.
$h = \frac{d_1}{4}$; $d_1 = 1.59\,d$	$V = \frac{\pi d_1^3}{16} = \frac{\pi d^3}{4}$	$S = 1.89\,\pi d^2$
$h = \frac{d_1}{2}$; $d_1 = 1.44\,d$	$V = \frac{\pi d_1^3}{12} = \frac{\pi d^3}{4}$	$S = 1.555\,\pi d^2$
$h = \frac{d_1}{3}$; $d_1 = 1.44\,d$	$V = \frac{\pi d_1^3}{12} = \frac{\pi d^3}{4}$	$S = 1.72\,\pi d^2$
$h = \frac{d_1}{2}$; $d_1 = 1.26\,d$	$V = \frac{\pi d_1^3}{8} = \frac{\pi d^3}{4}$	$S = 1.587\,\pi d^2$
$h = d$; d	$V = \frac{\pi d^3}{4}$	$S = 1.5\,\pi d^2$
$h = 1\frac{1}{2} d_1$; $d_1 = 0.87\,d$	$V = \frac{3}{8}\,\frac{\pi d_1^3}{4} = \frac{\pi d^3}{4}$	$S = 1.52\,\pi d^2$
$h = 2 d_1$; [illegible]	$V = \frac{1}{2}\,\frac{\pi d_1^3}{4} = \frac{\pi d^3}{4}$	$S = 1.57\,\pi d^2$

used. Local conditions, such as the position of valve-seats, etc., determine, of course, to a great extent, the form that must actually be given to the combustion space, but the fact remains, that it is well to confine the mean height of the combustion space to one not much less than one-third of the cylinder-diameter.

Let the mean height of the combustion space be denoted by h and let

V = the total cylinder volume, in cubic inches,
$n\,V$ = the volume of the combustion space,
d = the diameter of the cylinder, in inches,
and l = the length of stroke, in inches,

then
$$n\,V = \frac{\pi\,d^2}{4} \times h,$$
$$h = \frac{4\,n\,V}{\pi\,d^2}$$
and
$$d = \sqrt{\frac{4\,n\,V}{\pi\,h}}.$$

The total cylinder volume is
$$V = n\,V + \frac{\pi\,d^2}{4} \times l,$$
wherefore,
$$l = \frac{1-n}{n}\,h,$$
and
$$\frac{l}{d} = \frac{1-n}{n}\,\frac{h}{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (89)$$

By solving this equation for different values of n the following table will be obtained:

Value of $\frac{l}{d}$.

n	$\frac{h}{d} = 0.3$	$\frac{1}{3}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{2}$
$\frac{1}{2}$					1.00	1.5
$\frac{1}{3}$			1.00	1.5	2.00	
$\frac{1}{4}$		1.00	1.5	2.25		
$\frac{1}{5}$	1.2	1.33	2.00			
$\frac{1}{6}$	1.5	1.66	2.5			
$\frac{1}{7}$	1.8	2.00				
$\frac{1}{8}$	2.1	2.33				

Deciding on a ratio $\frac{h}{d}$ of about ⅓ as being favorable with respect to a limited cooling-surface, it becomes evident from this table, that for medium and low compressions a piston-travel of 1 to 1.33 times the cylinder-diameter will be ample; whereas, for the highest usual compression pressures the length of the stroke should approach 2 times the cylinder-diameter.

CHAPTER V

FUELS, COMBUSTION

CARBON and hydrogen and many combinations between these elements are ordinarily considered as fuels. Some fuels contain, in combination with carbon or hydrocarbon, an essential percentage of oxygen, but such are only the artificial fuels. Natural fuels, except when combined in the absorbed moisture, never contain any considerable percentage of oxygen.

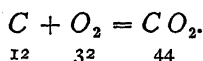
The principal natural fuels are: the various kinds of coal, wood, lignite, peat, coal-oils and natural gas. Among artificial fuels we have : producer gas, blast-furnace and coke-oven gas, city illuminating gas, alcohol, calcium carbide, etc. Producer gas is of particular interest relatively to the gas-engine, and its production and composition will therefore be given special attention in a following chapter.

Combustion.—The process of combustion consists in uniting chemically the elements of a fuel with oxygen, with the object in view to generate heat or light. Combustion is spoken of as being complete or incomplete.

In any furnace it is found that when an insufficient supply of air is furnished for the combustion of carbon less heat will be generated than when the proper allowance of air is supplied. This is due to incomplete combustion of part of the fuel, it being combined with only one-half of its proper allowance of oxygen to form a combustible gas, carbon monoxide. The combustion of the fuel to carbon monoxide gas generates in the furnace less than one-third of the full heating-value of the fuel, and the gas, thus formed, can, at any stage after it has left the furnace, be made to again combine with oxygen to the same amount as that which it already contains. It will, in doing this, generate more than twice the heat developed at its formation.

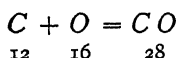
Reactions at the Combustion of Carbon.—The quantities of

the elements carbon (C) and oxygen (O) combining, at a complete combustion, to carbon dioxide (CO_2) are shown by the chemical formula expressing the reaction taking place. Thus,

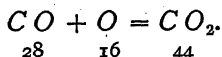
$$\underset{12}{C} + \underset{32}{O_2} = \underset{44}{CO_2}.$$

The figures shown below the symbols for the carbon and oxygen are, respectively, the atomic and the molecular weights of these elements, and they denote the weight-proportions in which the combination between the elements takes place. Further, the sum of the weights of the elements is the molecular weight of the result of the combination between them.

It appears, therefore, from the formula, that 12 parts of carbon, by weight, unite with 32 parts of oxygen to form 44 parts carbon dioxide gas, or, in other words, 1 pound of carbon unites, at complete combustion, with $2\frac{2}{3}$ pounds of oxygen to $3\frac{2}{3}$ pounds of carbon dioxide. During this combustion the full heating-value of the fuel is developed, or about 14,600 heat-units per pound carbon consumed.

The reaction at incomplete combustion of carbon is:

$$\underset{12}{C} + \underset{16}{O} = \underset{28}{CO}$$

That is, at incomplete combustion of one pound of carbon to $\frac{28}{12}$ ($= 2\frac{1}{3}$) pounds of carbon monoxide gas $\frac{16}{12}$ ($= 1\frac{1}{3}$) pounds of oxygen is consumed. The heating-value developed at this reaction is about 4,380 heat-units.

The reaction taking place when carbon monoxide gas burns to carbon dioxide is:

$$\underset{28}{CO} + \underset{16}{O} = \underset{44}{CO_2}.$$

One pound of carbon monoxide gas requires, thus, 0.572 pound of oxygen for the formation of 1.572 pounds of carbon dioxide. In this reaction there are generated again 4,380 heat-units. Hence, the $2\frac{1}{3}$ pounds of carbon monoxide gas that are formed from one pound of carbon require for combustion $2\frac{1}{3} \times 0.572$ ($= 1\frac{1}{3}$) pound of oxygen, or the additional amount that the original pound of carbon would have taken up had it, in the first

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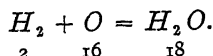
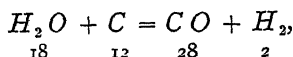
place, been burned to carbon dioxide; and the heat generated at the combustion of 2⅓ pounds of carbon monoxide is 2⅓ × 4,380 = 10,220 heat-units.

It will be seen, therefore, that carbon monoxide gas is a fuel-gas that contains a little more than two-thirds of the heating-value of the carbon from which it has been formed.

Reaction at the Combustion of Hydrogen—Dissociation.—The chemical reaction at the combustion of hydrogen is:

$$\underset{2}{H_2} + \underset{16}{O} = \underset{18}{H_2O}.$$

The formula shows that one pound of hydrogen (H) requires 8 pounds of oxygen (O) to combine into 9 pounds of water (H_2O).

This process may be reversed by carrying water-vapor through a hot porous coal-bed, whereby the water becomes dissociated into hydrogen and oxygen. The free oxygen will, however, not remain as such, but will instantly combine with carbon to form carbon monoxide gas, and the reaction becomes:

$$\underset{18}{H_2O} + \underset{12}{C} = \underset{28}{CO} + \underset{2}{H_2},$$

showing that 1½ pounds of water and one pound of carbon unite to 2⅓ pounds of carbon monoxide, and leave ⅙ pound free hydrogen.

In the combustion of hydrogen heat is generated to the amount of 62,100 heat-units per pound, but, on the other hand, when water is decomposed 62,100 heat-units are absorbed per pound of hydrogen formed, and must be furnished in order to maintain the reaction. In other words, for the decomposition of each pound of water a heat-supply of 6,900 heat-units is required.

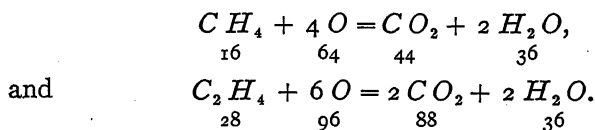
Hydrocarbons.—When coal, particularly bituminous coal, is heated in a retort there will be vaporized, or distilled off, a combustible gas consisting of hydrocarbons of varying compositions. The more important of these are:

methane (marsh-gas) CH_4,
and ethylene (olefiant-gas) C_2H_4.

The reactions taking place at their combustion are

$$\underset{16}{CH_4} + \underset{64}{4\,O} = \underset{44}{CO_2} + \underset{36}{2\,H_2O},$$

and

$$\underset{28}{C_2H_4} + \underset{96}{6\,O} = \underset{88}{2\,CO_2} + \underset{36}{2\,H_2O}.$$

The former gas requires $\frac{64}{16} = 4$ times its weight of oxygen for combustion, and the latter $\frac{96}{28} = 3.43$ times its weight.

It is often the case that the carbon and hydrogen contents in a hydrocarbon gas are given by the gas analysis, in per cent of the total weight of the gas. Thus, methane is composed of 75 per cent carbon and 25 per cent hydrogen, which, expressed in decimal proportions of the entire weight of the gas, is 0.75 and 0.25. In the preceding we have seen that each pound of carbon-element requires 2.667 pounds of oxygen for its complete oxidization, and each pound of hydrogen requires 8 pounds of oxygen.

The oxygen required for the combustion of methane-gas, CH_4, can, therefore, be figured as follows:

Oxygen required for the combustion of one pound CH_4.

by the carbon-element,		by the hydrogen-element,
0.75×2.667	+	$0.25 \times 8 =$
= (2 pounds)	+	(2 pounds) = 4 pounds O.

A general formula giving the oxygen required for the combustion of one pound of any hydrocarbon is, accordingly:

O required for combustion $= 2.667\,C + 8\,H$ pounds.

C and H expressing the proportions of carbon and hydrogen entering into the composition.

The Atmospheric Air.—Oxygen for the combustion of fuels is derived generally from the atmospheric air, which is composed of a mixture of oxygen and nitrogen in a proportion that may vary to some very slight extent. As average values of the proportion in which these elements enter in the atmosphere the following may be adopted:

	By Weight.		By Volume.
Oxygen,	0.231 parts	Oxygen,	0.21
Nitrogen,	0.769 parts	Nitrogen,	0.79
	1.000		1.00

According to these figures, it follows that:

One pound of oxygen is contained in $\frac{1,000}{231} = 4.33$ lbs. of air;

One pound of oxygen is associated with $\frac{769}{231} = 3.33$ lbs. of nitrogen;

One pound of nitrogen is contained in $\frac{1,000}{769} = 1.3$ lbs. of air;

One pound of nitrogen is associated with $\frac{231}{769} = 0.3$ lbs. of oxygen.

Weight of Air Required for Combustion.—One pound of carbon, requiring for complete combustion to CO_2-gas 2.667 pounds of oxygen, will thus consume $2.667 \times 4.33 = 11.554$ pounds of air, or, when burning to CO-gas, it will consume one-half of this amount, 5.777 pounds of air.

Likewise, one pound of CO-gas, requiring for its combustion to CO_2-gas 0.572 pound of oxygen, will consume $0.572 \times 4.33 = 2.477$ pounds of air, and one pound of hydrogen, requiring for combustion 8 pounds of oxygen, will consume $8 \times 4.33 = 34.64$ pounds of air.

The air required for complete combustion of methane-gas (consisting of 75 per cent C and 25 per cent H) can, therefore, be figured by the following method:

Air required for the combustion of one pound CH_4,

by the carbon-element,		by the hydrogen-element
0.75×11.554	+	$0.25 \times 34.64 =$
= (8.66 pounds)	+	(8.66 pounds) = 17.32 pounds.

A general formula giving the weight of air required for complete combustion of any hydrocarbon, whose weight-percentages of carbon and hydrogen are given by analyses, may be written:

Air required for combustion $= 11.554\ C + 34.64\ H$, or
Air required for combustion $= 11.554\ (C + 3\ H)$ pounds (90)

At the combustion of alcohol fuels, which already contain oxygen, there will be required, as will be shown later, less oxygen than what the carbon and hydrogen elements call for. This is due to the fact that the oxygen already in the combintion will, at

combustion re-combine with hydrogen to the amount required to form water. In estimating the air required for the combustion, allowance should, therefore, be made for the oxygen contained in the fuel by deducting a corresponding amount of air. The formula will be:

$$\text{Air required for combustion} = 11.554\,(C + 3\,H) - 4.33\,O.$$

This formula will, of course, apply also with reference to any hydrocarbon, because when there is no oxygen present in the fuel then the last term becomes zero.

Volume of Air Required for Combustion.—One pound of air occupying a volume of 12.39 cubic feet at 32° F. and a volume of 13.14 cubic feet at 62° F. the volume air required for complete combustion of the unit weight of a hydrocarbon will be:

$$\text{Vol. air at } 32^\circ\text{ F.} = 143\,(C + 3\,H) - 53.69\,O, \qquad (91a)$$
$$\text{Vol. air at } 62^\circ\text{ F.} = 152\,(C + 3\,H) - 56.85\,O, \qquad (91b)$$

or, in general, when the air is of a temperature t° F.

$$\text{Vol. of air at } t^\circ\text{ F.} = (0.29\,(C + 3\,H) - 0.109\,O)\,(460 + t) \qquad (91c)$$

Quantity of Air Required for the Combustion of a Compound Gas.—An average composition of anthracite producer gas is:

		Per cent Weight.	Per cent Volume.
Carbon monoxide	CO	27.6	26.0
Hydrogen	H	0.7	9.0
Methane	CH_4	1.2	2.0
Nitrogen	N	60.5	57.0
Carbon dioxide	CO_2	10	6.0
		100.0	100.0

Let it be required to find:

1. The weight of air necessary for complete combustion of one pound of the mixture, according to the weight percentages.

2. The volume of air, at 62° F., necessary for complete combustion of one cubic foot of the mixture, according to percentages of volume.

In Table X, page 109, there will be found:

The weight of air required for the combustion of one pound of each of the components contained in the mixture, also

The volume of air required for the combustion of one cubic foot of each of the components.

Accordingly we find:

The weight of air required for the combustion of 100 pounds of the gas mixture.

	Per cent Weight.	Air required for Comb. of 1 lb.	Total Wei of Air.
CO	27.6	2.477	68.37
H	0.7	34.64	24.25
CH_4	1.2	17.32	20.78
N	60.5	0.	0.
CO_2	10.0	0.	0.
			113.30

Per pound of gas there is required 1.133 pound of air.

The volume of air required for the combustion of 100 cubic feet of gas.

	Vol. of Components in 100 c.f.	C.f. of Air Req'd. for Comb. of 1 c.f. Gas.	Total Volume of Air Req'd.
CO	26.0	2.4	62.4
H	9.0	2.4	21.6
CH_4	2.0	9.61	19.22
N	57.0	0	0
CO_2	6.0	0	0
			103.22

Per cubic foot of gas there is required 1.03 cubic foot of air.

The Volume of Combustion-Products.—The weight of combustion-products is, of course, the same as the weight of the fuel plus the weight of air consumed.

To determine the total volume of the combustion-products, it will be necessary to settle on a certain temperature and pressure at which the determination shall be made, and, as the specific volumes of fuel-gases are, for the convenience of comparison, generally stated at a standard temperature and pressure, which is at 62° F. and one atmosphere, we may compute the volume of the combustion-products on the same basis.

At 62° F. and one atmosphere's pressure,

> CO_2-gas occupies 8.63 cubic feet per pound,
> and N-gas occupies 13.56 cubic feet per pound.

The H_2O-gas formed at the combustion of hydrogen would condense, at atmospheric pressure, before the low temperature 62° F. were reached. Its volume at 212° F. being, however, 26.36 cubic feet, hence, if for the sake of uniformity in the computations it be treated as a perfect gas, its volume at 62° F. becomes 20.45 cubic feet per pound.

Accordingly, the total volume of the products of combustion of one pound of methane gas, at standard temperature and pressure will be:

$$\begin{aligned}
\text{Vol. } CO_2 &= 0.75\ (1 + 2.667)\ 8.63 = & 23.73 \\
\text{Vol. } H_2O &= 0.25\ (1 + 8)\ 20.45 = & 46.01 \\
\text{Vol. } N &= (0.75 \times 2.667 + 0.25 \times 8)\ 13.56 \times 3.33 = & 180.62 \\
& \qquad\qquad\qquad\qquad\qquad \text{Total,} & 250.36
\end{aligned}$$

cubic feet at 62° F.

A general formula from which the combustion-products from any hydrocarbon may be computed, would be:

$$\text{Vol. at } 62^\circ \text{ F.} = \underset{\text{Vol. } CO_2.}{31.65\,C} + \underset{\text{Vol. } H_2O.}{184.05\,H} + \underset{\text{Vol. } N.}{120.41\,(C + 3\,H)} \qquad (92)$$

The volume at atmospheric pressure and at a temperature t° F., which is higher than 212° F., may be obtained by the formula

$$\text{Vol. at } t^\circ \text{ F.} = \frac{460 + t}{460 + 62} \times \text{Vol. at } 62^\circ \text{ F.}$$

At any temperature below 212° the volume of the H_2O-gas would contract, through condensation, to $\frac{0.0016}{20.45}$ of the volume computed at 62° F., or to practically nothing.

The above computations for the volumes of the combustion-products have all been made with the supposition that just enough oxygen, or air, has been utilized for the complete combustion of the fuel.

Calorific Power of Fuels.—At the combustion of one pound of hydrogen, 9 pounds of water are formed, and there is generated

heat to the amount of 62,100 heat-units. The latter is the calorific power of the gas as found by calorimetric tests at a constant pressure. This heat is all, in the first hand, absorbed by the products of combustion, and it is charged to them as sensible heat, and as latent heat of vaporization in the steam formed at the combustion; also as sensible heat in the admixed nitrogen, if the oxygen for combustion were derived from the atmosphere.

In the calorimeter, the total heating-value of a gas is obtained by condensing and cooling its products of combustion to a temperature near 32° F., or to the temperature at which the gas is supplied, and determining the heat thereby abstracted. If, in absorbing the heat from the combustion-products, we cannot bring their temperature below 212° F., it is evident that the latent heat in the steam admixed will not become available, and that, therefore, at any temperature above 212 degrees, the actual heating-value realized will be unaffected by the total amount of the latent heat in the products of combustion. Similarly, when the heating-value of a gas is utilized, as in a gas-engine, for increasing the volume of the working charge, it is evident that the increase in volume, due to the latent heat of vaporization, will not be available for power, unless the temperature, in cooling, is brought below 212 degrees. However, such a low temperature, at the end of the expansion in the gas-engine, is not obtainable in practice. When the lower practical temperature-limit is above 212 degrees, we should, therefore, subtract the latent heat in the products of combustion from the total heating-value of the fuel-gas, in order to obtain its "effective heating-value." This effective heating-value of a gas is referred to as its "lower calorific value."

On the other hand, when the temperature, in cooling, is brought as low as 32° F., then the full heating-value of the gas becomes available, and this value is referred to as the "higher calorific value" of the gas, and it is, as stated, for hydrogen 62,100 B. T. U.

Any gas containing hydrogen, free, or in combination with carbon, will, therefore, have a higher and a lower calorific value, and the latter is always obtained from the former by subtracting

the latent heat in the water of combustion, or that part of the heat of combustion that has no bearing on the efficiency of a motor utilizing the heat.

The heat of vaporization of water, from and at the temperature of 32° F., may be obtained from Regnault's formula,

$$H = 1091.7 - 0.695\ (t - 32),$$

by giving the temperature of vaporization, t, a value $t = 32°$.

Thus, $H = 1091.7$ B. T. U.

As one pound of hydrogen forms, at combustion, nine pounds of water, the heat of vaporization absorbed by the water, at the combustion of each pound of hydrogen, is $9 \times 1091.7 = 9{,}825$ heat-units. This heat deducted from the higher calorific value, 62,100 heat-units, gives 52,275 heat-units as the lower calorific value of the gas.

If, in any hydrocarbon, the percentage of hydrogen expressed in decimals be x, the vapor formed at the combustion of the gas becomes $x \times 9$ pounds, and the heat of vaporization will be $x \times 9 \times 1{,}091.7$. Therefore, $x \times 9{,}825$ heat-units should be subtracted from the higher calorific value of the gas in order to obtain its lower value.

Calorimetry.—Calorimeters, in general, are arranged for burning in the apparatus, with ample supply of air or oxygen for complete combustion, a determined quantity of the fuel to be tested, and for absorbing the heat generated by means of a measured quantity of water. If the increase in the temperature of the water be carefully observed, the heat evolved at the combustion per pound of fuel will be obtained directly from the equation

$$Q = \frac{W\ (t_2 - t_1)}{G}; \quad . \quad . \quad . \quad . \quad . \quad (93)$$

when W is the weight of the cooling water, $t_2 - t_1$ its increase in temperature, and G the weight of the fuel consumed.

The Junker Gas Calorimeter.—The most generally used apparatus for determining the heating-value of a fuel-gas is the Junker gas-calorimeter, shown in Fig. 33, by which a continuous fuel-test may be obtained. In connection with the apparatus there are

used the gas-meter *M* and a pressure-regulator *R;* the latter consisting, simply, of an ordinary water-sealed gas-bell by which the gas-pressure is maintained constant.

The gas to be tested is admitted to the meter by means of a rubber tube connection at *g*, and after being metered it is conducted through the pressure-regulator to the burner *B*, which is

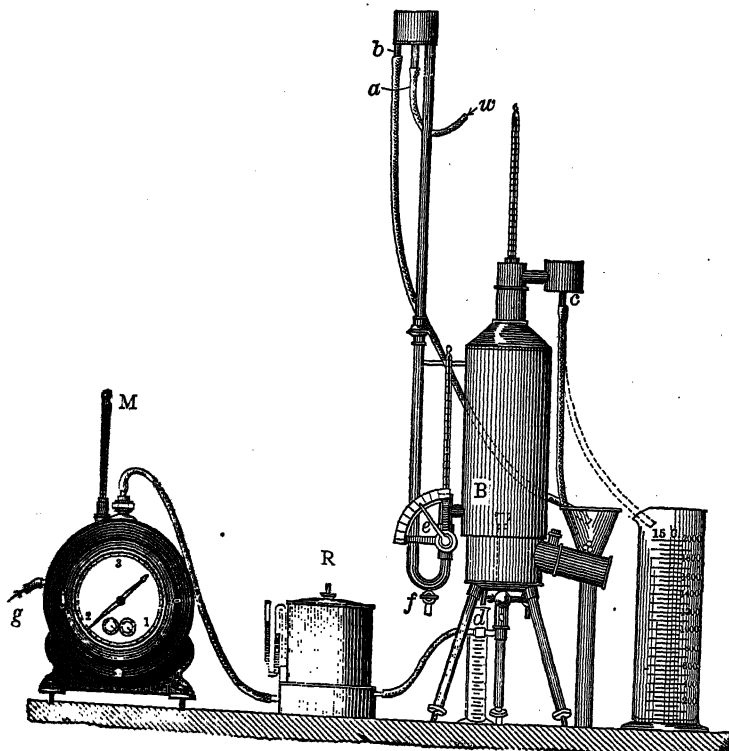



FIG. 33.

shown in dotted outlines at the lower part of the calorimeter. The combustion-products pass to the upper part of the combustion-chamber, and from there through double rows of vertical tubes surrounding the combustion-chamber, from which the gases are delivered to the uptake-chamber and exit-flue at the base of the chamber. While passing out, the gases are thoroughly cooled

and condensed by the cooling water which surrounds the tubes and the combustion-chamber.

The cooling water enters the small constant-level chamber at "*a*," from which it flows, at a constant head, to the apparatus, entering it through a regulating cock *e* and thermometer pocket; *b* is an overflow-pipe carrying away any surplus water delivered to the chamber.

The temperature of the cooling water is measured by the thermometers inserted in pockets provided at the inlet to and at the discharge from the apparatus. The water is discharged through a small free-level chamber *c*, which serves to obviate any siphoning action that would be caused by the vertical discharge tube, and it is either emptied in the waste funnel, during the period of starting for a test, or to the measuring glass, during the run of the test. The condensation, which results from the combustion of hydrogen or compounds of hydrogen, will collect at the bottom of the uptake-chamber, and is drained into a separate graduated measuring glass *d*. The apparatus is thoroughly jacketed so as to prevent as much as possible any heat-loss through radiation.

Preliminary to a test the apparatus must be in operation for some time, until the condensation begins to flow at a normal rate, after which the test may be started, by reading the meter, shifting the cooling water and condensation discharge to the collecting beakers, and observing the thermometers. The combustion is regulated so as to admit an ample amount of air, yet not too great an excess. The rate at which producer gas is consumed in the apparatus is normally about six cubic feet per hour.

The result from a test-run, of any duration, may be obtained directly from the readings during corresponding time-limits.

The measuring glasses belonging to the apparatus are generally graduated in litres (one litre being the equivalent of one kilogram of pure water) and the thermometers in centigrades. The result, the weight of the cooling water in kilograms times the increase in its temperature in centigrades will, therefore, be expressed in calories, which are the equivalent of 3.968 B. T. U.

If V is the volume of gas consumed, in cubic feet, reduced to atmospheric pressure and 62° F.,
W the weight of the cooling water, in kilograms,
t_2, centigrade, the mean discharge temperature of the cooling water, and
t_1, centigrade, the mean inlet temperature of the cooling water,
then the calorific power of the gas per cubic foot will be

$$H = 3.968 \frac{W (t_2 - t_1)}{V} \text{ B. T. U.}$$

The weight of hydrogen contained in the fuel is one-ninth the weight of the water of condensation collected.

The Mahler Calorimeter.—For solid or for liquid fuels the Mahler bomb calorimeter, Fig. 34, is often used, and it is generally considered a correct and reliable apparatus for the determination of the heating-value of such fuels.

It consists simply of a small steel vessel, the bomb, in which a predetermined quantity of the fuel is burned in an atmosphere of compressed oxygen, while the instrument is submerged into a measured quantity of water that absorbs the heat of combustion.

The steel bomb is of only a small capacity, and as even a very small sample of fuel consumes a large quantity of oxygen, it is necessary that the oxygen shall be of a considerable pressure in order that the fuel shall become completely consumed. In practice, a pressure of 300 pounds is ordinarily used. In order, therefore, that it shall withstand the high pressure, the bomb must be of considerable strength, and, as iron or steel is readily oxidized at high temperatures, it must be lined on the inside by a material that will be unaffected by the action of oxygen, or acids formed at the combustion.

A usual arrangement of the apparatus is shown in Fig. 34. B is the porcelain-lined steel bomb having suspended from the stopper a small platinum pan C on which the fuel-sample for combustion is placed—if a solid fuel, it is pulverized before being weighed out and charged. After being sealed, the apparatus is slowly charged with oxygen of the required pressure, through a small valve in the stopper, and immersed into the

body of water, D, whose weight is known and whose temperature is carefully read before and after the combustion. The fuel is ignited by means of an electric battery, P, of high tension; the leads from which are connected with the bomb and with the insulated contact-pole E, leading from the ignition-wire, at F, and passing through the cover.

The lever L operates a system of vanes for the agitation of the water in the vessel D while the combustion takes place, so that its average temperature shall be registered by the thermometers immersed in that vessel. The increase in the temperature of the water in the outside vessel, A, due to heat transmitted through

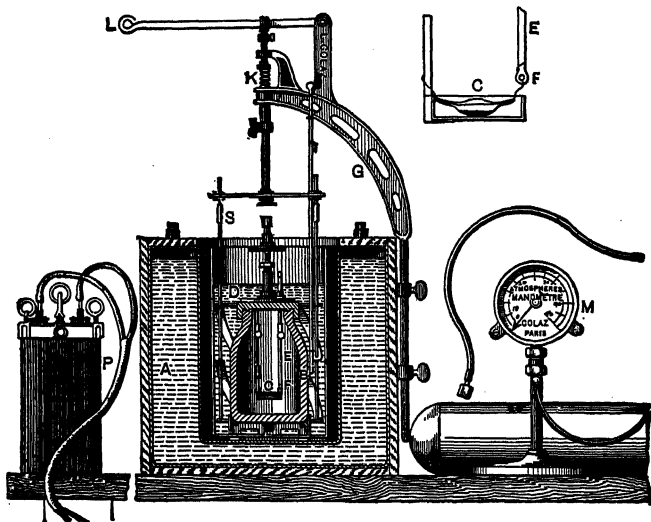



FIG. 34.

the insulating air-body between the inside and outside vessels, is also registered by thermometers reading to within a small fraction of one degree, and the final result is corrected for the heat corresponding to this increase in temperature.

In order to give a correct result, this apparatus must be skilfully handled, and carefully calibrated for the heat absorbed by the metallic parts of the apparatus itself. This heat is, per unit weight of the metallic parts, so much less than that absorbed by

the water as the average specific heat of the metal is less. The weight of a quantity of water having the same heat-absorbing capacity as the apparatus is determined and added to the actual weight of the water involved, for being inserted in place of W in formula 93.

Carefully conducted calorimeter tests of various fuels have been made by many experimenters, of which those by Berthelot and Scheurer-Kestner are now most generally quoted. According to the former, the calorific value of hydrogen is 62,100 B. T. U. and that of carbon 14,600 B. T. U.

These fundamental values have been quoted in the tables, pages 108 and 109, and will be used for computations in the following.

Formulas for the Calorific Value of Fuels.—A chemical analysis of a fuel is, under many circumstances, more readily accomplished than a calorimetric test, and it becomes quite possible to determine, approximately, the heating-value of a fuel from its analysis.

A well-known, generally accepted, approximate formula for the calorific value of a fuel is of the following form:

$$Q = 14{,}600\, C + 62{,}000 \left(H - \frac{O}{8}\right) \quad . \quad . \quad . \quad (94)$$

Q being the calorific power, per pound,
C the percentage carbon,
H the percentage hydrogen,
and O the percentage oxygen.

C, H, and O to be expressed in decimal parts of the total weight of the fuel as unit.

The formula is based on two important laws by the physicist Dulong:

The heating-value of a compound fuel is the sum of the heating-values that would be obtained by the combustion of each component part separately, providing that the fuel does not contain oxygen and hydrogen chemically combined with it.

When oxygen and hydrogen exist in chemical combination with a fuel, it is only that part of the hydrogen outside of what is required for forming water with the combined oxygen that

will evolve heat; and the oxygen required for combustion will be so much less than what the fuel-analysis would show as the amount of oxygen that the fuel already contains.

Accordingly, as 8 pounds of oxygen are required for one of hydrogen, the weight of hydrogen has, in the formula, been reduced with ⅛ of the weight of the combined oxygen, and the surplus only counted as actual fuel-value. It will be noticed, further, that the formula is based on a heating-value of 14,600 B. T. U. for carbon and 62,000 B. T. U. for hydrogen.

For liquid hydrocarbon fuels the following formula gives results closer to those of the calorimeter, and is frequently employed in practice.

$$Q = 14{,}500\,C + 52{,}230\left(H - \frac{O}{8}\right) \quad . \quad . \quad (94a)$$

For fuels containing an appreciable amount of sulphur, which at combustion adds to the heating-value of the fuel, the following formula will give a close approximation·

$$Q = 14{,}500\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}050\,S \quad . \quad (94b)$$

S being the decimal part of sulphur contents in the fuel.

Diversities of opinion are expressed as to which of the two calorific values of a gas should be the proper one to charge against the performance of an engine. At present, however, when there are utilized so many fuels of different composition, the question should be considered from two viewpoints. When the object is to consider the economy of the gas-engine in general, in comparison with other machines for the same purpose, viz., for converting heat-energy into power, then the engine should be charged up with the total heat-energy with which it is supplied, and this is the high calorific value of the gas. This seems fair, because, on any other basis, the disadvantage of the gas-engine, in not being able to utilize heat to the same low temperature as, for instance, the condensing steam-engine, would not be brought out. When, on the other hand, the performances of two gas-engines on different fuels are compared, then the fact should not be lost sight of that one engine may be charged with

a fuel, as, for instance, hydrogen, the heating-value of which is less available than that of the fuel, for instance *CO*-gas, with which the other engine may be charged. In this case the lower calorific value should be the standard for comparison.

The lower calorific value of hydrogen is namely fully 15 per cent lower than its high calorific value, whereas in the case of CO-gas the high and the low calorific values are the same.

The American Society of Mechanical Engineers' code of standards for engine tests recommends the higher calorific value to be used. In reporting the economy obtained by an engine under test, it is important, however, that there should be stated whether the economy has been based on the higher or lower heating-value of the fuel. If this information is furnished, as well as an analysis of the fuel, there can then always be made a fair comparison between the engine under test and any other on a different fuel.

As far as the heating-value of a fuel-gas is concerned, it is evident that it is only the temperature and the specific heat that have influence on the efficiency with which the gases do their work. But the fact remains that, other things being equal, the less the heating-value that is spent as latent heat in the gases of combustion the higher the efficiency will be.

Specific Heat.—The specific heat of a substance is the quantity of heat required to raise the temperature of the unit weight of it one degree.

The quantity of heat that must be supplied in order to increase the temperature of a gas one degree depends on the conditions under which the gas is heated; whether it is allowed to increase its volume freely when being heated, or whether it is confined in a certain space, in which latter case the pressure to which the gas is subjected increases as heat is added. Gases have, therefore, two different specific heats, one at a constant pressure and one at a constant volume. For ordinary temperatures the specific heats of gases have been found to be practically constant, but they cannot be said to be so for temperatures obtaining in furnaces.

Flame-Temperature.—The maximum temperature of the products due to a combustion of a fuel, the so-called flame-

temperature, may be approximately computed on the assumption that the specific heat of the combustion-products is constant for all temperatures, up to the maximum obtained.

TABLE VI.

Specific Heat of Gases at 212° F. Mainly according to Regnault.

NAME	Symbol	At constant pressure C_p	At constant volume C_v	$\frac{C_p}{C_v}$
Air		0.2374	0.1700	1.41
Superheated steam	H_2O	0.4805	0.3585	1.34
Hydrogen	H	3.4090	2.4177	1.41
Oxygen	O	0.2175	0.1543	1.41
Nitrogen	N	0.2438	0.1729	1.41
Carbon monoxide	Co	0.2479	0.1738	1.41
Carbon dioxide	Co_2	0.2169	0.1570	1.29
Marsh gas	CH_4	0.5929	0.4505	1.316
Olefiant gas	C_2H_4	0.4040	0.3200	1.26
Natural gas				
Average anthracite gas		0.24 to 0.26		
Average bituminous gas ...		0.24 to 0.26		
Average blast-furnace gas .		0.23		
Average chimney-waste gas		0.24		

It is evident that the heating-value evolved must be absorbed primarily by the products of combustion, hence, if Q be the lower heating-value of a fuel-gas, W the weight of its combustion-products, and C the mean specific heat of the latter, then the flame-temperature becomes

$$T = \frac{Q}{CW}.$$

T is, of course, actually, the increase in temperature of the combustion-products from the initial, but, as the final temperature generally is high and the formula only approximate, it will be practically correct to call T the flame-temperature.

The products of a combustion consist generally of air, nitrogen, CO_2-gas and steam, and, when the weight of each of these elements present in the combustion-products is known, their mean specific heat is readily computed.

Let it be required to find the approximate flame-temperature of a fuel-gas of a heating-value of 2,000 B. T. U. per pound; the combustion-products from one pound of the gas being of the following composition:

	Combustion-Products. lbs.		Specific Heat at Constant Press.		
Air	0.16	×	0.24	=	0.0384
N.....................	1.48	×	0.24	=	0.3552
CO_2	0.57	×	0.22	=	0.1254
H_2O (superheated steam)	0.09	×	0.48	=	0.0432
	2.30				0.5622

Mean specific heat of all the combustion-products

$$= \frac{0.5622}{2.3} = 0.244.$$

The heating-value of one pound of the fuel-gas being 2,000 B. T. U., the flame-temperature becomes

$$T = \frac{2,000}{2.3 \times 0.244} = 3,570° \text{ F.}$$

The heat evolved during a combustion at a constant pressure is, practically, the same as that evolved if the volume remains constant. Hence, as of the two specific heats that at a constant pressure is the highest, the flame-temperature due to a combustion under a constant pressure must be less than that due to a combustion at a constant volume.

If in the preceding computation the specific heats at constant volume are used, we get:

The mean specific heat of all combustion-products = 0.175, and the flame-temperature, when the volume of the gases remains constant

$$T = 5,000° \text{ F.}$$

When the combustion of a gas is carried out, as in practice, in a cylinder, there will be dissipated a considerable amount of heat by being conducted away into the metal of the cylinder, and the actual flame-temperature becomes materially lower than the theoretical figure.

The average theoretical flame-temperatures of various fuels burned in air are given in the following table.

Theoretical Flame-Temperatures.

Fuel.	Flame-Temperature degrees Fahr.
C burned to CO	2,600
C burned to CO_2	4,900
CO burned to CO_2	5,400
Hydrogen	4,800
Marsh gas	4,000
Producer gas	2,200
Coal gas	4,800

Density of Gases.—By Avogadro's law the simple fact is established that equal volumes of all gases, at the same temperature and pressure, contain the same number of molecules, and as a consequence it follows that the ratio between the weights of equal volumes, or between the weights of unit volumes, of any two gases are as the ratio between their molecular weights.

The molecular weights of substances are customarily referred to the weight of the hydrogen-molecule as unit; it being the lightest of them all. Thus, when the molecular weight of hydrogen is figured as 2, and that of any other given gas is called m, the ratio between the weight of unit volumes of hydrogen and the given gas will be as 2 to m, or as 1 to $\frac{m}{2}$.

Hence, the weight of the unit volume of a gas, or its density is:

$$\text{density of a gas} = \frac{m}{2} \times \text{density of hydrogen.}$$

To determine, therefore, the weight per cubic foot of any gas, it will be necessary only to establish a constant for the weight of one cubic foot of hydrogen, which, when multiplied by one-half the molecular weight of the gas, will give this quantity.*

* This operation is the reversal of the process by which the molecular weights have, in the first hand, been determined. However, the even numbers by which the molecular weights are generally expressed are not in all cases absolutely correct, wherefore a, with respect to fuel-gases, unimportant discrepancy may be involved in the results of the reversed computation.

Late determinations have shown the weight of hydrogen to be 0.005591 pound per cubic foot, at the temperature 32° F. and at the atmospheric pressure (14.7 pounds per square inch).

Hence, the density, W_{32}, of a gas, at 32° F. and at a pressure of one atmosphere is

$$W_{32} = 0.005591 \frac{m}{2}.$$

The inverted value of the figure 0.005591, or 178.87, is the number of cubic feet of hydrogen that will make up one pound, under the above specified temperature and pressure conditions.

Example.—The molecular weight of marsh gas, CH_4, being 16, the ratio of equal weights of hydrogen and marsh gas is, therefore, as 1 to $\frac{16}{2}$, and the weight of marsh gas, in pounds per cubic foot, at 32° F., and at one atmosphere's pressure, is

$$= 0.005591 \times 8 = 0.044728.$$

Likewise, the molecular weight of carbon-dioxide being 44, its weight per cubic foot at 32° F. and at the atmospheric pressure is

$$= 0.005591 \times \frac{44}{2} = 0.123.$$

The density of gases, at a constant pressure, varying as the inversed value of their absolute temperatures, the density, W_t, of a gas, at the temperature t, will be

$$W_t = \frac{459.2 + 32}{459.2 + t} W_{32}.$$

Hence, at the temperature 62° F., the temperature to which standard gas is referred, its density will be

$$W_{62} = \frac{491.2}{521.2} W_{32} = 0.94244\, W_{32}.$$

Thus, at standard temperature and pressure, the weight per cubic foot

of marsh gas $= 0.94244 \times 0.044728 = 0.04215$ pound.
of carbon-dioxide $= 0.94244 \times 0.123 = 0.1159$ pound.

It may sometimes be convenient to refer the density of a gas not to the density of hydrogen, but to that of air. As dry air of a temperature of 32° F., and of atmospheric pressure is $\frac{0.08728}{0.00559}$ (= 14.44) times the weight of hydrogen, therefore,

$$\text{density of a gas referred to dry air} = \frac{m}{28.88}; \quad . \quad (95)$$

when m is the molecular weight of the gas.

We have seen that:

$$\text{density of a gas} = \frac{m}{2} \, 0.005591$$

and this equation inverted becomes

$$\frac{1}{\text{density of a gas}} = \frac{357.74}{m}.$$

The molecular weights of all gases referring simply to the relative weights of equal volumes of the substances, it is inconsequential what volumes may be assumed to be involved in a volumetric computation. It sometimes becomes convenient to figure with a unit volume represented by the figure 357.74, which is twice the volume of the unit weight of hydrogen, at 32° F., and at one atmosphere's pressure. This volume is called the molecular volume of hydrogen, and it serves as a unit, at volumetric computations, just as ½ the molecular weight of hydrogen serves as a unit to which the molecular weights of substances are referred.

Calorific Value at Constant Pressure and at Constant Volume.— Some slight difference exists between the heating-value of a fuel, measured at constant pressure, and that obtained at a constant volume. This is due to the fact that the combustion-products from many gaseous fuels are, at equal temperature and pressure, less in volume than the gases consumed in the combustion. Hence, the atmosphere in compressing the material from the larger volume V_i, before combustion, to the smaller volume V_f, after combustion, is supplying it the heating-value

$$\frac{1}{J} (V_i - V_f) P;$$

P being the constant pressure at which the combustion takes place.

Assuming that one pound of fuel-gas, together with the oxygen required for complete combustion, is consumed. The volumes involved will then be:

The fuel-gas, $\frac{1}{a}$ molecular volumes

The oxygen, $\frac{1}{b}$ molecular volumes

The combustion-products, $\frac{1}{c}$ molecular volumes

and, hence,

$$(V_i - V_f) = 357 \cdot 74 \left(\frac{1}{a} + \frac{1}{b} - \frac{1}{c}\right).$$

The fractions $\frac{1}{a}$, $\frac{1}{b}$ and $\frac{1}{c}$ depend on the proportions in which the substances combine.

The volume of nitrogen involved in the combustion-process may be disregarded, because, referred to the temperature 32° F., it remains unchanged.

The difference between the heating-value, Q_p, at constant pressure and that, Q_v, at constant volume, will be

$$Q_p - Q_v = \frac{1}{J}(V_i - V_f) P$$

$$\frac{2116.3 \times 357 \cdot 74}{778}\left(\frac{1}{a} + \frac{1}{b} - \frac{1}{c}\right)$$

$$973\left(\frac{1}{a} + \frac{1}{b} - \frac{1}{c}\right).$$

Example.—For the combustion of 1 pound of hydrogen = ½ molecular volume there is required 8 pounds of oxygen = $\frac{8}{16} \times$ ½ = ¼ molecular volume, and the combustion-products will be 9 pounds = $\frac{9}{18}$ = ½ molecular volume.

The combustion-products (steam) will, however, at 32° F., be condensed into a volume practically = 0. We may say, therefore, that

$$Q_p - Q_v = 973 (\tfrac{1}{2} + \tfrac{1}{4} - 0)$$
$$= 973 \times \tfrac{3}{4}$$
$$= 729 \text{ B. T. U.}$$

Thus, the heating-value of hydrogen at constant pressure being 62,100 B. T. U., it becomes at constant volume 62,100 − 729 = 61,371 B. T. U.

The difference between the two heating-values being so small, it is not generally taken much account of in practice, but, in order, sometimes, to explain discrepancies in heating-values obtained by different experimenters, it is well to have the matter in view.

Heating-Value per Unit Volume.—In practice it is, of course, much more convenient to measure a large quantity of gas by its volume rather than by its weight, and the gas consumed is, on this account, generally debited to an engine under test in cubic feet. Instead of having the heating-value of a gas given in heat-units per pound it will, therefore, be convenient to have it expressed in heat-units per cubic foot. For a known gas this reduction is made by dividing its heating-value per pound by its density.

In order to compare the value of two volumes of gas it is, however, necessary that they should be of the same temperature and pressure, and, to make the comparison conveniently, the quality of a gas is customarily given at a temperature of 62° F., and at a pressure of 14.7 pounds, which is the temperature- and pressure-conditions of "standard gas."

To reduce the volume of a given quantity of gas, of a pressure p_i and temperature t_i, to a volume at standard temperature and pressure, the following formula will be convenient. Designate the initial condition of the gas as follows:

T_i = its absolute temperature = $t_i + 459.2°$ F.,

p_i = its absolute pressure = 14.7 + gauge-pressure,

V_i = its volume, in cubic feet;

and let

T_f = the absolute temperature of standard gas
= 62° + 459.2 = 521.2°,

p_f = the atmospheric pressure, at the sea-level, = 14.7 pounds.

V_f = the volume of the gas at standard temperature and pressure.

$$\text{Then,}\quad V_f = V_i \frac{p_i}{p_f} \frac{T_f}{T_i}$$

$$= V_i \frac{521.2}{14.7} \frac{p_i}{T_i} = 35.4\, V_i \frac{p_i}{T_i}.$$

Natural gas is generally sold at a gauge-pressure of 4 oz. = ¼ of one pound.

To reduce the volume of a quantity of gas of this pressure to a volume at standard pressure, its temperature being supposed to remain constant, we use the formula:

$$V_f = \frac{14.7 + 0.25}{14.7} V_{4\,oz}. \quad . \quad . \quad . \quad . \quad (96)$$

$$V_f = 1.017\ V_{4\,oz}.$$

Or, reversed, if it be required to find what volume a quantity of standard gas will have at a pressure of 4 oz., by the gauge, we write:

$$V_{4\,oz} = 0.983\ V_f. \quad . \quad . \quad . \quad . \quad . \quad . \quad (96a)$$

Specific Gravity.—The words specific gravity, specific weight, density and relative density are often used referring to the same thing. The most general use of the words are: Density to mean the weight of the unit volume of a substance, and

Specific gravity to mean the ratio of the density of a substance to that of some standard substance. The weights of solids and fluids are generally referred to the weight of equal volumes of water at 39.2° F., and gases to air or hydrogen as standards.

The density of fuel-oils is very often expressed by the number of degrees it corresponds to on the Baumé hydrometer-scale. The table on the following page will give the specific gravity corresponding to any given degree Baumé.

Vapor-Pressure.—Assuming the laws for perfect gases to hold good with reference to a fuel-vapor, then the weight of the vapor present in a unit volume, at a given constant temperature, is proportional to its vapor-pressure. In a condensible vapor there is a definite limit to the amount of vapor that can exist in a unit space, at any given temperature. Hence, there is for each given temperature of the vapor a limit to its vapor-pressure, or a limit

to the pressure that it can sustain without becoming partially condensed into liquid.

In raising the temperature of a vapor its maximum vapor-pressure, or vapor-pressure of saturation, is increased considerably

TABLE VII.

Baumé and Specific-Gravity Equivalents.

Degrees Baumé.	Specific Gravity.	Degrees Baumé.	Specific Gravity.	Degrees Baumé.	Specific Gravity.
10	1.0000	37	0.8395	64	0.7243
11	0.9930	38	0.8346	65	0.7205
12	0.9861	39	0.8299	66	0.7168
13	0.9791	40	0.8251	67	0.7133
14	0.9722	41	0.8204	68	0.7097
15	0.9658	42	0.8157	69	0.7061
16	0.9594	43	0.8110	70	0.7025
17	0.9530	44	0.8063	71	0.6990
18	0.9466	45	0.8017	72	0.6956
19	0.9402	46	0.7971	73	0.6923
20	0.9339	47	0.7927	74	0.6889
21	0.9280	48	0.7883	75	0.6856
22	0.9222	49	0.7838	76	0.6823
23	0.9163	50	0.7794	77	0.6789
24	0.9105	51	0.7752	78	0.6756
25	0.9047	52	0.7711	79	0.6722
26	0.8989	53	0.7670	80	0.6689
27	0.8930	54	0.7628	81	0.6656
28	0.8872	55	0.7587	82	0.6619
29	0.8814	56	0.7546	83	0.6583
30	0.8755	57	0.7508	84	0.6547
31	0.8702	58	0.7470	85	0.6511
32	0.8650	59	0.7432	86	0.6481
33	0.8597	60	0.7394	87	0.6451
34	0.8544	61	0.7357	88	0.6422
35	0.8492	62	0.7319	89	0.6392
36	0.8443	63	0.7281	90	0.6363

and for each pressure there is a certain temperature where partial condensation begins. This temperature is called the saturation temperature for the given pressure. If the temperature of a saturated vapor is lowered (heat abstracted) then the vapor be-

comes moist (not saturated) and if its temperature is raised (heat added) it becomes superheated.

In the following table is given the vapor-pressure of saturation of various fuel substances, and, for the purpose of comparison, that of water. The values for the alcohols are from the "Smithsonian Miscellaneous Tables," and the values for benzol, hexane, and the gasoline fuels are those obtained by Sorel,* at investigations for the purpose of establishing the relative values of the alcohol and the gasoline fuels. Sorel found that there exists no definite relation between the density and vapor-pressure of different commercial gasoline fuels; the composition of the fuels being too indefinite. The vapor-pressure of the more volatile products

TABLE VIII.

Vapor-Pressure of Saturation for Various Liquids.

Temperature.		Vapor-Pressure of Saturation, in Millimeters of Mercury.						
							Gasoline.	
°C.	°F.	Pure Ethyl Alcohol.	Pure Methyl Alcohol.	Benzol.	Water.	Hexane C_6H_{14}	Automobiline. Sp. Density at 62° F. 0.703.	Stelline. Sp. Density at 62° F. 0.673.
0	32	12	30	27	5	45	99	164
5	41	17	40	36	7	58	115	190
10	50	24	54	45	9	74	133	220
15	59	32	71	61	13	95	154	255
20	68	44	94	77	17	119	179	296
25	77	59	123	96	24	154	210	358
30	86	78	159	120	32	184	251	433
35	95	103	204	156	42	228	301	512
40	104	134	259	188	55	276	360	596
45	113	172	327	224	71	335	422	685
50	122	200	409	271	92	401	493	792
55	131	279	508	326	117	482	561	
60	140	350	624	390	149	567	648	
65	149	437	761	468	187	674	739	

* "Carburation et Combustion dans les Moteur à Alcool," par E. Sorel.

in the fuels is, however, considerably higher than that of hexane ($C_6 H_{14}$) which represents, fairly, the composition of the average gasoline fuel.

The vapor pressure of kerosene is, according to Boverton Redwood,* at 60° F., 1 $^m/_m$ mercury, thus considerably less than that of water.

The Pressure of a Gas Mixture.—When two gases enclosed in the same receptacle are subjected to an outside pressure, they both help to sustain that pressure in the ratio of their respective gas-pressures, and the sum of the pressures of the two gases must equal the total pressure to which they are subjected.

In a given mixture of air and a saturated fuel-vapor, which may be assumed to follow the laws of perfect gases, the ratio of

FIG. 34*a*.

$p_1 + p_2$

$v_1 + v_2$

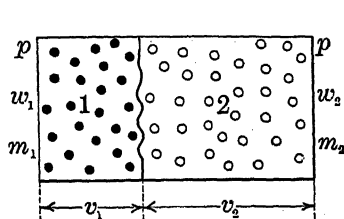
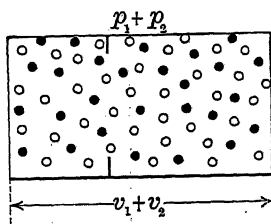



FIG. 34*b*.

the pressures respectively of the air and of the vapor is equal to the ratio of the volumes of the gases reduced to the same pressure. Or, in other words, the ratio of the weight of the air and of the vapor present in a given volume equals the direct ratio of their respective gas-pressures times their densities. These facts are readily derived on the basis of the illustrations Figs. 34*a* and 34*b*, and are perhaps most clearly expressed by the formulas given below.

Fig. 34*a* represents a vessel containing two perfect gases designated (1) and (2) which, it must be assumed, are chemically non-reactive upon each other. When held separated by a yielding partition, as represented in the figure, the pressure of both gases is the same, p; their volumes are respectively v_1 and v_2; their weights w_1 and w_2; and their densities or molecular weights

* "Petroleum and its Products," by Sir Boverton Redwood.

TABLE IX.

Data Pertaining to Elementary Fuels and Combustion-Products.

	Chemical Symbol	Molecular Weig't.	Per Cent of Each Element in the Compound, by Weight.		Volume in C. F. of One Pound of Gas at 14.7 Pounds Pressure per Sq. In. at		Weight per C. F. at 14.7 Pounds Pressure and at		Calorific Value of the Gas. In B. T. U. per pound Gas.		In B. T. U. per Cub. Ft. of Gas. High Value at		Low Value at	
			% C.	% H. or O.	32° F.	62° F.	32° F.	62° F.	High	Low.	32° F.	62° F.	32° F.	62° F.
Hydrogen	H_2	2			178.87	189.8	0.005591	0.005269	62100	52275	347	327	292	275
Methane (marsh gas)	CH_4	16	75	25	22.36	23.73	0.04473	0.04215	24000	21544	1073	1013	968	910
Ethylene (Olefiant gas)	C_2H_4	28	85.7	14.3	12.78	13.56	0.07827	0.07376	21900	20496	1714	1615	1600	1512
Acetylene	C_2H_2	26	92.31	7.69	13.74	14.60	0.07268	0.06849	21860	21094	1592	1500	1540	1440
Ethane	CH_6	30	80	20	11.92	12.65	0.08386	0.07903	22340	20375	1880	1800	1700	1620
Butylene	C_4H_8	56	85.7	14.3	6.39	6.78	0.15655	0.1476	20860	19405	3264	3080	3036	2860
Benzene	C_6H_6	78	92.31	7.69	4.59	4.87	0.21805	0.2055	18100	17343	3940	3720	3780	3560
Carbon monoxide	CO	28	42.86	57.14	12.78	13.56	0.07827	0.07376	4380	4380	344	324	344	324
Carbon	C_2	24	.....	.....	.....	.....	.....	.....	14600	14600	.....	.....	.....	.....
Carbon dioxide	CO_2	44	27.3	72.7	8.13	8.63	0.123	0.1159	.....	.....	.....	.....	.....	.....
Nitrogen	N_2	28	.....	.....	12.78	13.56	0.07827	0.07376	.....	.....	.....	.....	.....	.....
Oxygen	O_2	32	.....	.....	11.18	11.87	0.08936	0.08421	.....	.....	.....	.....	.....	.....

TABLE X.

Data Pertaining to Elementary Fuels and Combustion-Products.

	Chemical Symbol.	Molecular Weig't.	Chemical Reaction at Combustion.	Req'd Weight of: O. per Lb. Gas.	Req'd Weight of: Air, per Lb. Gas.	Weight of Combustion-Products per Pound Gas.			Volume of Combustion-Products per Pound Gas. at 32° Fahr.			Cub. Ft. of Air Req'd for Combustion.	
				Lbs.	Lbs.	CO_2 Lbs.	H_2O Lbs.	N. Lbs.	CO_2 Cub. Ft.	H_2O. Cub. Ft.	N. Cub. Ft.	Per Lb. Gas at 62° F.	Per C. F. Gas.
Hydrogen	H_2	2	$2H_2+O_2=2H_2O$	8	34.64	......	9	26. 64	......	178.87	340.46	456	2.4
Methane (Marsh gas)	CH_4	16	$CH_4+2O_2=CO_2+2H_2O$	4	17.32	2.75	2.25	13. 32	22.36	44.72	170.23	228	9.61
Ethylene (Olefiant gas)	C_2H_4	28	$C_2H_4+3O_2=2CO_2+2H_2O$	3.4285	14.845	3.143	1.286	11.417	25.55	25.58	145.91	195	14.3
Acetylene	C_2H_2	26	$2C_2H_2+5O_2=4CO_2+2H_2O$	3.077	13.323	3.385	0.692	10.246	27.52	13.75	130.94	175	12.0
Ethane	C_2H_6	30	$2C_2H_6+7O_2=4CO_2+6H_2O$	3.733	16.164	2.933	1.8	12.431	23.85	35.77	158.87	213	16.83
Butylene	C_4H_8	56	$C_4H_8+6O_2=4CO_2+4H_2O$	3.4285	14.845	3.143	1.286	11.417	25.55	25.58	145.91	195	28.8
Benzene	C_6H_6	78	$2C_6H_6+15O_2=12CO_2+6H_2O$	3.077	13.323	3.385	0.692	10.246	27.52	13.75	130.94	175	36.0
Carbon monoxide	CO	28	$2CO+O_2=2CO_2$	0.572	2.477	1.572	......	1.905	12.78	......	24.35	33	2.4
Carbon	C_2	24	$C+O_2=CO_2$	2.6667	11.555	3.6667	......	8.888	29.81	......	113.59	152	......

m_1 and m_2. Assume now the two gases to be mixed as in Fig. 34*b*, then their gas-pressures become changed, say, to p_1 and p_2 so that $p = p_1 + p_2$, and we have

$$\frac{p_1}{p} = \frac{v_1}{v_1+v_2} \text{ and } \frac{p_2}{p} = \frac{v_2}{v_1+v_2}, \quad \text{hence } \frac{p_1}{p_2} = \frac{v_1}{v_2}.$$

The rel. volume of the gas (1) is $\frac{w_1}{m_1}$, and that of the gas (2) $\frac{w_2}{m_2}$,

$$\text{thus} \quad \frac{v_1}{v_2} = \frac{w_1\, m_1}{w_2\, m_2}, \qquad \text{and hence } \frac{w_1}{w_2} = \frac{p_1\, m_1}{p_2\, m_2}.$$

The gas designated (1) may be assumed to represent a saturated fuel-vapor of a vapor-pressure p_1 at a given temperature, while (2) represents air of the same temperature and of a barometric pressure p_2, and the preceding formulas become then expressions for the relative volumes, or weights, of vapor and air in any given fuel and air mixture.

When the vapor-pressure of a fuel is, under ordinary atmospheric conditions, much weaker than that of the air, it may happen that the proportion of fuel to that of air in a saturated mixture is too small for forming a good explosive mixture, or the mixture may be too weak in fuel even to be explosive. If such is the case, then, in order to obtain a combustible mixture, the fuel must be heated until its vapor-pressure becomes high enough to form a proper mixture. The question of the minimum temperature at which a properly vaporized mixture can be obtained becomes of special interest with respect to the alcohol-fuels, which, at ordinary temperatures, are of rather weak vapor-pressures.

In carrying out computations relating to the properties of the commonly used fuel-gases under different conditions, reference must often be made to data concerning the constituent elementary gases, and it becomes, therefore, an object to have such data arranged in the most convenient manner, in tabular form, for ready reference. Tables IX and X on the preceding pages have been computed on the basis of the established weight of hydrogen, 0.005591 pound per cubic foot, at the temperature of freezing water, 32° F., and at a pressure of 14.7 pounds per square inch; and the heat-values are those determined by Berthelot.

CHAPTER VI

GAS-ENGINE FUELS—THE PROPORTIONING OF THE MIXTURES AND THE RELATION OF THESE TO THE SIZE OF THE ENGINE

The Density of a Charge after Completed Suction-Stroke.—The maximum power of a given gas-engine depends largely on the properties of the fuel with which it is charged, and for estimating its average power-capacity it is necessary to determine the maximum heating-value it can take in, to advantage, per suction-stroke, or per minute. It is evident that the greater the density of a given explosive mixture the more heating-value it will contain, and, as its density depends on its temperature and pressure, the cooler the mixture is, and the higher its pressure after completed suction-stroke, the more heating-value the cylinder will contain for transformation.

When, during the forward motion of the piston, the mixture is drawn in to the cylinder, the pressure existing there must, of course, be below the pressure of the gas-supply, and it remains slightly below that pressure up to the time the compression begins on the return stroke. The less the difference is between the supply pressure and the pressure in the cylinder after completed suction-stroke the greater, therefore, will the power-capacity of the engine be.

With ample valve-areas, and on a full load, when there is no wire-drawing of the charge by a governor throttling-valve, the loss in density of the gas-mixture, due to reduced pressure, should not be more than that corresponding to a pressure-reduction of about 1½ pounds.

In case the gas is supplied to the engine at atmospheric pressure, the suction pressure, p_a, would, accordingly, be 13.2 pounds and the corresponding density of the gas, therefore,

$$\frac{p_a}{p_o} d_o = \frac{13.2}{14.7} d_o = 0.9\, d_o,$$

when d_o is its density at atmospheric pressure, p_o.

While expanding from the pressure p_o to p_a, the temperature of the gas-mixture will be reduced to the extent that

$$T_a = \left(\frac{p_a}{p_o}\right)^{0.3} T_o;$$

T_o and T_a being the absolute temperatures of the mixture, before, and after the expansion. This equation, solved for different ratios $\frac{p_a}{p_o}$, gives the following values for T_a and corresponding temperature-reduction $T_o - T_a$, assuming T_o to be $460 + 62°$ F.

$p_o - p_a$	$\frac{p_a}{p_o}$	T_a	$T_o - T_a$
½ pound	0.966	517	5°
1	0.932	511	11°
1½	0.899	506	16°
2	0.865	500	22°

Accordingly, assuming that the gas and air arrive to the engine at atmospheric pressure and of a temperature 62° F., and assuming a pressure-reduction in the cylinder of 1½ pound, then the temperature of the charge, when inside the cylinder, would be 62 − 16 = 46° F.

There are, however, two main sources that contribute to the heating of the charge during its admission to the cylinder. These are: heat absorbed from the exhaust valve, piston, and walls of the combustion-chamber and cylinder, and heat contained in the exhaust gases remaining in the cylinder from the preceding stroke. Of least influence is probably the heat of the exhaust gases (neutrals) with which the new charge is mixed.

For the purpose of an approximate estimate of the maximum heat that, in an average case, can be transmitted to the incoming charge by the cylinder-walls, we assume a 24 × 36 cylinder whose walls are of the temperatures shown in Fig. 35. These temperatures are, of course, assumed to be at the inside surface of the material only, particularly when the outside surface is in contact with cooling water. The surface temperatures of the piston and exhaust valve, if uncooled, as well as of the main part

of the combustion-chamber, being quite high, they may in the estimate be allowed a value 800° F.

For the dimensions given, the wall-surfaces of the cylinder are approximately:

Cylinder barrel	19 square ft.,	of a temperature of 400°F.
Comb. chamber and piston	8 square ft.,	of a temperature of 800°F.
Total	27 square feet;	

and the average temperature per square foot surface, accordingly, 510° F.

The charge arriving to the cylinder at a temperature of 62 − 16 = 46° F., and becoming heated to, say, 74°, the mean temperature-difference will be 510° − 60 = 450° F.

The maximum amount of heat the interior surface of the

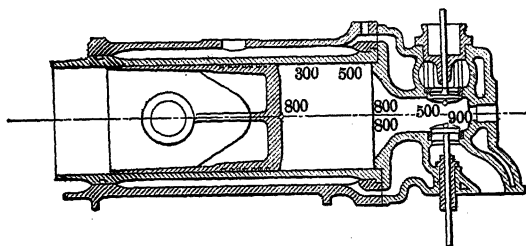



FIG. 35.

cylinder can be assumed to transmit to the incoming charge, per square foot surface, per hour and degree temperature-difference, is 8* heat units; taking into account the fact that the gas will be in lively circulation. The total maximum amount of heat transmitted per hour by 27 square feet interior surface, therefore:

$$27 \times 450 \times 8 = 97{,}200 \text{ H.U. per hour.}$$

The charge absorbing heat from the cylinder only during, say, ¼ of the time lapsing, it will receive only ¼ × 97,200 = 24,300 H.U. per hour.

The weight of the charge passing through the cylinder is, at a piston-speed of 600 feet per minute,

* Approximate estimate according to Dulong-Petit and Péclet's laws for radiation and convection.

$$= \frac{60 \times 600 \times 3.14}{4 \times 13.14} = 2,150 \text{ pounds per hour.}$$

13.14 cubic feet of the charge being the equivalent of one pound.

Hence, the heat abstracted from the cylinder-walls per pound of the charge becomes

$$\frac{24,300}{2,150} = \text{approximately } 11.3 \text{ H.U.}$$

At a piston speed of 600 feet per minute, we may say, therefore, that the temperature of the mixture will, during the suction stroke, become increased a maximum of 46° F. The piston and exhaust valve being assumed not to be water-cooled. With water-cooled piston and exhaust valve its temperature will be increased approximately only 26° F.

These figures may be considered as maxima, and at a higher rate of speed the heat-absorbtion should be somewhat less.

The influence of the neutral gases on the temperature of the charge would, for a general case, be estimated as follows:

Let by *gas-mixture* be understood the new charge of air and gas admitted to the cylinder, and let by *normal charge* be understood the gas-mixture together with the proportion of neutrals remaining in the combustion-chamber from the preceding stroke.

Assuming that the gas-mixture arrives to the inlet valve at a temperature of 62° F. and at atmospheric pressure, it has been shown that its temperature, then, through expansion in entering the cylinder, will be reduced to 62° − 16 = 46° F., and it will again, through absorbtion of heat from the cylinder-walls, be raised a maximum of 46°; when exhaust valve and piston are not water-cooled, and 26° if these parts are cooled.

When mingling with the neutrals the temperature of the gas-mixture is, therefore,

46 + 46 = 92° F., when exhaust valve and piston are not water-cooled, and 46 + 26 = 72° F., when exhaust valve and piston are water-cooled.

The temperature of the neutrals will vary quite extensively, but under proper conditions it should not exceed 220°,* at atmos-

* R. H. Fernald, Transact. Am. Soc. Mech. Engs., Vol. XXIII.

pheric pressure; this temperature corresponding to a temperature at release of about 1,800° F.

The ratio between the total cylinder-volume and the clearance-volume being as r to 1, hence a volume $= r - 1$ gas-mixture mingles with a volume = 1 neutrals. These volumes of gas, however, being of different temperatures, they must be reduced to a common temperature in order to be comparable as to weights and densities.

We have:

The absolute temperature of the gas-mixture =

$$T_m = 460 + 92 = 552.$$

The absolute temperature of the neutrals =

$$T_n = 460 + 220 = 680.$$

The absolute temperature corresponding to 62° F. =

$$T_o = 460 + 62 = 522.$$

Therefore, the volume of the gas-mixture at 62° F. =

$$\frac{T_o}{T_m}(r - 1)$$

and the volume of the neutrals at 62° F. =

$$\frac{T_o}{T_n}(1).$$

At equal pressures, the specific weights of the gas-mixture and of the neutrals are equal, and equal to that of air, which is $\frac{1}{13.14}$ at 62° F.

Further, the specific heat at constant pressure of these gases can be assumed to be the same, $= c_p$.

The weight of the new gas-mixture admitted to the cylinder is, therefore,

$$W_m = \frac{1}{13.14}\frac{T_o}{T_m}(r - 1),$$

and the weight of the corresponding proportion neutrals is

$$W_n = \frac{1}{13.14}\frac{T_o}{T_n}.$$

If T_x signifies the common temperature of the gas-mixture and neutrals mingled, we have

$$c_p (T_x - T_m) \frac{1}{13.14} \frac{T_o}{T_m} (r - 1) = c_p (T_n - T_x) \frac{1}{13.14} \frac{T_o}{T_n}$$

$$\therefore T_x = \frac{r T_m}{(r-1) + \dfrac{T_m}{T_n}}. \quad . \quad . \quad . \quad . \quad (97)$$

Computing, from this formula, the temperature of the charge for two cases, respectively, when the exhaust valve and piston are not water-cooled, and when these parts are water-cooled, and for alternately two extreme compression ratios, $r = 4$ and $r = 7$, we assume as given the following:

(1) $r = 4$, $T_n = 680$, $T_m^{\text{I}} = 532$, water-cooled.
$T_m^{\text{II}} = 552$, non-water-cooled.

(2) $r = 7$, $T_n = 680$, $T_m^{\text{III}} = 532$, water-cooled.
$T_m^{\text{IV}} = 552$, non-water-cooled.

The following temperatures will be obtained:*

	Water-cooled exhaust valve and piston.	Non-water-cooled exhaust valve and piston.
= 4,	$t_x^{\text{I}} = 103°$ F.	$t_x^{\text{II}} = 120°$ F.
$r = 7$,	$t_x^{\text{II}} = 89°$ F.	$t_x^{\text{IV}} = 107°$ F.

The preceding computation would not hold good if the neutrals were of a slight pressure when the exhaust valve closes, but in most cases, with mechanically moved valves, it is perfectly possible so to adjust the opening of the inlet valve and the closing of the exhaust valve, that the actual amount of neutrals remaining shall not exceed that which the assumption of fully expanded neutrals would contemplate. It is feasible even, with suitable adjustment of the valves, to obtain still further scavenging.

The Heating-Value of the Expanded Normal Charge.—When the mixture is delivered to the engine at standard temperature and pressure, and when there is a drop in pressure of 1½ pound at the admission of the charge to the cylinder, we see, thus, that the

* The temperatures that actually will be obtained depend, of course, largely on how perfectly the valves are adjusted for releasing the discharge effectively. With good automatic scavenging the temperatures should, however, approach those given.

maximum temperature of the charge after completed suction-stroke, will be approximately 120° F., as a maximum.

The ratio between the specific volume of the mixture after completed suction-stroke and its specific volume at standard pressure and temperature is

$$\frac{V_a}{V_o} = \frac{P_o}{P_a}\frac{T_a}{T_o} = \frac{14.7}{13.2}\frac{580}{522} = 1.11 \times 1.11 = 1.23.$$

The volume of the expanded gas-mixture, per cubic foot of fuel-gas, after completed suction-stroke is, therefore,

$$\frac{V_a}{V_o}(x\,a + 1) = 1.23\,(x\,a + 1);$$

$x\,a + 1$ being the volume of the cold gas-mixture, per cubic foot of fuel-gas, as it is delivered to the engine.

If H be the heating-value per cubic foot of fuel-gas, the quotient

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)}$$

is the heating-value per cubic foot of the expanded normal charge, after completed suction-stroke.

The value of this factor is what most directly determines the power-capacity of an engine of given cylinder-volume, and it may be estimated for any fuel-gas of known properties as follows:

Displacement Required per Horse-power.—It is to be assumed that the chemical analysis of the fuel-gas is known, and from it may be computed, according to equation 91*b*, the volume of air required for the complete combustion of the gas. Allowing $x - 1$ per cent excess air, above what the analysis calls for, we obtain:

The volume of air required according to analysis	a.
The volume of air required practically . . .	$x\,a$.
The volume of the explosive mixture, cold . .	$x\,a + 1$.
The volume of the expanded final charge, after completed suction-stroke	$\frac{V_a}{V_o}(x\,a + 1)$
The heating-value per cubic foot of the expanded normal charge	$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)}$

The minimum suction-displacement necessary per indicated horse-power

$$D = \frac{\frac{42.42}{H}}{Efy\,\frac{V_a}{V_o}(xa + 1)} \quad . \quad . \quad . \quad . \quad . \quad (52)$$

By means of Table I, the approximate value of the efficiency Efy, can be determined when the compression suitable for the gas is known. This determination calls, however, for some previous experience with the gas in question, as well as with the type and construction of the engine. Different fuels allow of vastly different intensity of compression, and it has been seen, that the same fuel may be compressed, without bad effects, to a much higher pressure in one type of engine than in another. This is due to the more or less successful construction of the combustion-chamber and ignition device.

The minimum suction-displacement required for the number of horse-power that are to be provided for can conveniently be solved from equation 52.

It must be observed, however, that the capacity of the engine to generate the required power, even at a slight overload, must be considered. That is, the engine must have some overload capacity. It is customary, therefore, to rate the engine at a somewhat less power than its maximum, to allow for short heavy impulses in the resistance. The suction-displacement per rated indicated horse-power should on this account, according to good practice, be made at least 15 per cent larger than that given by equation 52.

An engine is also rated by its brake horse-power, which is the power that can be taken off from the engine shaft by means of a brake and brake-wheel. The brake horse-power is, thus, the indicated horse-power less the power consumed in friction in the engine itself.

On an average, it may be said that the friction in a gas-engine amounts to 15 per cent of its full power, and the remaining 85 per cent can be obtained as brake horse-power.

Hence we may say that, on an average, the maximum I.H.P. 1.15 rated I.H.P. $= \frac{1}{0.85}$ maximum B.H.P. $= \frac{1.15}{0.85}$ rated B.H.P.

According to these figures, the required suction-displacement, in cubic feet, per minute, will be:

$$D_1 = \frac{42.42}{E f y \frac{H}{\frac{V_a}{V_o}(x a + 1)}} \text{ per maximum I.H.P.} \quad (52)$$

$$D_2 = 1.15 \frac{42.42}{E f y \frac{H}{\frac{V_a}{V_o}(x a + 1)}} \text{ per rated I.H.P.} \quad (52a)$$

$$D_3 = 1.17 \frac{42.42}{E f y \frac{H}{\frac{V_a}{V_o}(x a + 1)}} \text{ per maximum B.H.P.} \quad (52b)$$

$$D_4 = 1.35 \frac{42.42}{E f y \frac{H}{\frac{V_a}{V_o}(x a + 1)}} \text{ per rated B.H.P.} \quad (52c)$$

It will be evident that

$$m.e.p. \times D_1 = \frac{33{,}000}{144} = 229.2;$$

$m.e.p.$ being the mean effective pressure per square inch of the piston.

We get, therefore, the relation between the mean effective pressure and the suction-volume required per minute per indicated horse-power.

$$D_1 = \frac{229.2}{m.e.p.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (98)$$

The following table gives the required displacement-volume per horse-power, corresponding to various mean effective pressures and for different horse-power ratings. In the last column are each of the commonly used gas-engine fuels placed opposite

the displacement-volume per horse-power per minute that conservatively will be required for it.

TABLE XI.

M.E.P. and Suction-Displacement per Horse-Power.

Normal M. E. P.	Normal Suction-Displacement, Cubic Feet.				Suction-Displacement Volume Required for Various Fuels.
	Per Max. I. H. P. D_1	Per Rated I. H. P. $D_2 = 1.15\ D_1$	Per Max. B. H. P. $D_3 = 1.176\ D_1$	Per Rated B. H. P. $D_4 = 1.35\ D_1$	
60	3.82	4.39	4.49	5.16	
62	3.70	4.25	4.35	5.00	
64	3.58	4.12	4.21	4.83	
66	3.47	3.99	4.08	4.68	
68	3.37	3.88	3.96	4.55	
70	3.27	3.76	3.85	4.41	Anthracite Suction Gas
72	3.18	3.66	3.74	4.29	
74	3.10	3.56	3.65	4.19	Coke-Oven Gas
76	3.02	3.47	3.55	4.08	Bituminous Gas
78	2.94	3.38	3.47	3.98	
80	2.87	3.30	3.38	3.87	
82	2.80	3.22	3.29	3.77	Illuminating Gas
84	2.73	3.14	3.21	3.68	Kerosene
86	2.67	3.07	3.14	3.60	
88	2.61	3.00	3.07	3.52	Natural Gas
90	2.55	2.93	3.00	3.44	Gasoline
92	2.49	2.87	3.93	3.36	
94	2.44	2.81	2.87	3.29	
96	2.39	2.75	2.81	3.23	
98	2.34	2.69	2.75	3.16	Alcohol
100	2.292	2.64	2.70	3.09	
102	2.25	2.59	2.65	3.04	
104	2.20	2.53	2.60	2.98	
106	2.16	2.48	2.55	2.92	
108	2.12	2.43	2.50	2.86	
110	2.08	2.38	2.45	2.81	

For a single-acting one-cylinder four-cycle engine, there is one suction-stroke for every two revolutions; therefore, if N be the number of revolutions per minute, a the area of the piston in square inches, and l the length of the stroke in inches, then we have

$$\frac{l\,a\,N}{2 \times 12 \times 144} \qquad D_2\,.\,\text{I.H.P.} = D_4.\,\text{B.H.P.}$$

Hence,

$$\text{I.H.P.} = \frac{1}{D_2}\,\frac{l\,a\,N}{3{,}456} \text{ and} \quad . \quad . \quad (99a)$$

$$\text{B.H.P.} = \frac{1}{D_4}\,\frac{l\,a\,N}{3{,}456}. \quad . \quad . \quad . \quad . \quad (99b)$$

The mean effective pressure corresponding to the heating-value, $\dfrac{H}{\frac{V_a}{V_o}(x\,a + 1)}$, per cubic foot of the final charge will be

$$m.e.p. = 5.4\,E\,f\,y\,\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} \quad . \quad . \quad (50b)$$

The Normal Charge in Hit-or-Miss Engines.—When it is desired to determine the volume and temperature of the final charge in a hit-or-miss engine, the following deductions will be useful:

Let V_g be the volume of gas consumed per minute,
and T_1 its absolute temperature;
· V_a the volume air consumed per minute,
and T_2 its absolute temperature;
N the number of revolutions of the engine, per minute,
and x the number of explosions per minute;
then $\frac{1}{2} N$ is the number of suction-strokes,
and $\frac{1}{2} N - x$ the number of misses per minute.

The volume of gas per explosion-stroke is $\dfrac{V_g}{x}$, the volume of air per explosion-stroke is

$$\frac{V_a - \frac{V_g}{x}\left(\frac{1}{2} N - x\right)}{\frac{1}{2} N} = \frac{2\,(V_a + V_g)}{N} - \frac{V_g}{x}$$

and the volume of gas-and-air mixture is $\dfrac{2\,(V_a + V_g)}{N}$.

At the temperature T_m—the temperature obtained at the mixture of the gas and air—the volumes are:
The volume gas consumed per explosion-stroke

$$V_1 = \frac{T_m}{T_1}\,\frac{V_g}{x},$$

The volume air consumed per explosion-stroke

$$V_2 \quad \frac{T_m}{T_2}\left(\frac{2\,(V_a + V_g)}{N} - \frac{V_g}{x}\right),$$

and the volume of the mixture $= V_1 + V_2$.

A volume $(r - 1)$ gas-and-air mixture, of the temperature T_m, mingles, in the cylinder, with a volume 1 neutrals, of the temperature T_n.

Hence, the volume of the neutrals of the absolute temperature T_n is $\frac{1}{r-1}(V_1 + V_2)$, and the volume of the neutrals of the absolute temperature T_m is $\frac{T_m}{T_n}\,\frac{1}{r-1}(V_1 + V_2)$.

If it be assumed that the air, gas, and neutrals are of the same specific gravity and specific heat, and if T_x designates the temperature of the final mixture, we have,

$$V_1\,(T_x - T_m) + V_2\,(T_x - T_m) = \frac{T_m}{T_n}\,\frac{1}{r-1}(V_1 + V_2)\;(T_n - T_x);$$

thus
$$T_x - T_m = \frac{T_m}{T_n}\,\frac{1}{r-1}\,(T_n - T_x),$$

or
$$T_x = \frac{r\,T_m}{(r-1) + \dfrac{T_m}{T_n}}.$$

It can generally be assumed that the air and gas are of the same temperature when being mixed, thus, $T_m = T_1 = T_2$, if not, we solve the equation

$$\frac{V_g}{x}(T_m - T_1) = \left(\frac{2\,(V_a + V_g)}{N} - \frac{V_g}{x}\right)\;(T_2 - T_m),$$

and obtain $T_m = T_2 - (T_2 - T_1)\,\dfrac{N\,V_g}{2\,(V_a - V_g)\,x}$.

Petroleum Fuels.—In refining raw mineral oil (petroleum), there are produced fuel-oils of different densities and volatility, and, in the way of by-products, mineral lubricating oils. The lighter products, petroleum ether, gasoline and benzine, vaporize and form with the atmosphere very readily explosive mixtures, and they are on this account not safe to store or to use in quantities in closed rooms.

In order that a hydrocarbon vapor and air shall become explosive there must be present in the mixture the required amount of fuel-gas; if the vapor is too highly diluted with air no explosion can take place. It has been seen in the previous chapter, that the amount of vapor that can vaporize from a liquid, and mix with the air, at a certain temperature, depends on the vapor-pressure of the gas at the given temperature, and hence it is evident that hydrocarbons of low enough vapor-tension cannot form explosive mixtures at ordinary atmospheric conditions, unless the temperature be increased to the point at which their vapor-tension becomes high enough to allow the required proportion gas to form and mix with the air.

The degree of safety a fuel-oil offers against fire-risk is measured by its flashing-point, which is the temperature to which the fuel must be raised in order to give an explosive mixture, when under atmospheric pressure. The burning test of kerosene, it is required, shall be such, that the first flash shall be obtained at a temperature not lower than 110 degrees Fahrenheit.

The refining process results in a variety of products, and it can be conducted so as to, at will, produce a greater or less proportion of oils of a high volatility. Through the so-called cracking-process, or destructive distillation, in refining oils a considerable quantity of the oil of a composition intermediate between kerosene and lubricating oil is converted into hydrocarbons of lower density and boiling-point, and thus made suitable for fuel or illuminating purposes. The products of the fractional distillation are, therefore, not identical with the hydrocarbons present in the crude oil, but the result of this treatment is that the comparatively heavy oils undergo dissociation, and the yield of kerosene becomes increased. The object is generally to reduce the main bulk of the fuel-oils into kerosene, which is a product that can be transported and utilized with comparative safety.

The table on the following page shows the average proportions of the products obtained.

Gasoline.—Gasoline is the most volatile of the fuel-oils resulting from the refining of crude petroleum, and it is itself composed of a number of hydrocarbons of varying densities and boiling-

TABLE XII.

Products Obtained at the Refining of Petroleum.

Average Values.

Temperature Degrees Fahr.	Distillate.	Per Cent.	Specific Gravity.	Density, Baumé	Boiling-Point, Fahr.	Flashing-Point.
113 to 140	Rhigolene, Chymogene } Petroleum ether	traces	.590 to .625	85–80	104–158	....
140 to 158	Gasoline	1.5	.660 to .670	80–78	158–176	....
158 to 248	Benzine naphtha C	10.	.680 to .700	78–68	176–212	14
....	Benzine naphtha B	2.5	.714 to .718	68–64	212–248	....
....	Benzine naphtha A	2	.725 to .737	64–60	248–302	32
248 to 347	Polishing-oils	......	....	......	....	....
338+	Kerosene.............	50	.753 to .864	56–32	302–572	100–122
482	Lubricating oils	15	.864 to .960	32–15	572–up	230
....	Paraffine wax.........	2	....	......	....	....
....	Residuum and loss.....	16	....	......	....	....

points. The average density of the fuel as found in the markets is now rarely less than 79° Baumé and more commonly nearer 62°, or from 0.672 to 0.732 specific-gravity.

As an average analysis of the gasoline used for motor purposes may be taken the following:

Carbon	83.5 per cent.
Hydrogen	15.5 per cent.
Nitrogen, Sulphur, Oxygen, etc.,	1.0 per cent.
	100.0

The average composition of gasoline is therefore approximately expressed by the formula $C_6 H_{14}$.

Its calorific value varies:

The high value, from 19,000 to 21,000 B.T.U. per pound; the lower value, from 17,500 to 19,500 B.T.U. per pound.

The weight of air required for its combustion is, per pound: weight air required $= 11.548 (0.835 + 3 \times 0.155) = 15$ pounds, approximately, or counted in cubic feet, at the temperature 62° F., and at atmospheric pressure, the required volume becomes; volume air required $= 152 (0.835 + 3 \times 0.155) = 198$ cubic feet.

Vaporized Gasoline as Fuel.—The higher calorific value of gasoline varies from 19,000 to 21,000 B.T.U. per pound, and at the combustion of each pound of fuel there is formed, on an average, 1.4 pound of water which absorbs about 1,500 heat-units for its vaporization into dry steam.

The lower calorific value is, therefore, using a rather conservative figure, 18,500 B.T.U. per pound.

The volume of one pound of gasoline vapor is, at the atmospheric pressure and 62° F.,* 4.2 cubic feet, and according to analysis there is required for its complete combustion 15 pounds, or 198 cubic feet of air per pound of the fuel.

For complete combustion in the gas-engine cylinder some margin of excess air, above what the analysis calls for, is required, and if this margin be made 15 per cent. excess there would be required, practically, $15 + 2.25 = 17.25$ pounds. The total weight of the mixture from one pound of fuel becomes, thus, 18.25 pounds.

From Table VIII of pressures of saturation of gasoline, page 106, it will appear that at the temperature of 86° F., its pressure is 251 $^m/_m$ mercury. The pressure of the atmosphere is, on an average, approximately, three times this pressure, but, on the other hand, the weight of the gasoline vapor is approximately three times as heavy as the air, and, hence, at the temperature of 86° F., a saturated mixture of air and gasoline contains equal weights of air and gas. Such a mixture is too rich in fuel to be explosive, but if the air is charged with less than one-quarter of the fuel required for saturation, and until it contains only a small percentage of fuel, it will be explosive. That a saturated mixture of air and fuel is non-explosive is illustrated by the experiment, which can be made, with due precaution, of striking fire to a

* Assuming the gasoline to be of a composition corresponding to the formula C_6H_{14}, then, according to equation 95, page 101, the density of the gas becomes $\frac{86}{28.88}$ =, approximately, three times that of air. Its volume per pound, therefore, $\frac{13.14}{3} = 4.36$ cubic feet at 62° F. For average gasoline fuels the figure 4.2 will answer, corresponding to a molecular weight of 89. Compare with experiments made by A. H. Gill and H. R. Healey, Technology Quarterly, 1902, Vol. XV, page 74.

match in a vessel containing gasoline in which the air is fully saturated with gas. Under these conditions the fuel-mixture will not burn, whereas if the match be lighted several feet away from the vessel containing the fuel, where the air may be charged with only a small suitable quantity of fuel, an explosion will result. On account of the high vapor-pressure of the fuel, and since a very small quantity of fuel is required, an explosive mixture is very readily formed, under all atmospheric conditions, if a current of air be passed over the surface of the fuel-liquid.

The effect of the vaporization of gasoline in the carbureter is to lower the temperature of the air-and-gas mixture and thereby to increase the density with which it enters the cylinder. This factor can readily be taken account of in the computation for the heating-value per unit volume of the charge.

The latent heat of gasoline is, approximately, 180 B.T.U., and there are 18.25 pounds of mixture that must supply, in the main, the heat demanded for the vaporization of each pound of fuel. Hence, each pound of the mixture must supply approximately 10 heat-units, and in doing this its temperature will be reduced approximately 40 degrees Fahrenheit; the specific heat of the air and vapor mixture being 0.25. Evidence of the material lowering of the temperature of the charge is found in the frost often forming on the carbureter.

In the general estimate for the temperature of the final charge after completed suction-stroke, it was assumed that the gas and air arrived to the engine at a temperature of 62° F. In the case of gasoline this figure must be reduced to approximately $62 - 40 = 22°$ F., in order to obtain the average temperature at which the carbureted gasoline-mixture is supplied.

The temperature of the final gasoline-charge after completed suction-stroke, computed as before, but starting from the temperature 22° F., instead of 62° F., will be about 86° F. The ratio, therefore, between the specific volume of the final charge after completed suction-stroke and its specific volume at standard temperature and pressure will be

$$\frac{V_a}{V_o} = \frac{P_o}{P_a} \cdot \frac{T_a}{T_o} = \frac{14.7}{13.2} \cdot \frac{546}{522} = 1.15.$$

The drop in pressure during the admission to the cylinder being allowed, as before, 1½ pound.

The temperature-changes to which the charge is subjected during its admission to the cylinder will be explained by the following schedule:

The gasoline and air arrive to the carbureter at the temperature 62° F. The temperature of the mixture after carburation = 62 − 40 = 22° F.; after expansion at entering the cylinder = 22 − 16 = 6° F.; after being heated by the cylinder = 6 + 46 = 52° F.; final charge after mingling with the neutrals = 86° F.

The estimate of the heating-value per cubic feet of suitable mixture will appear as follows:

The volume of air required by analysis per cubic foot of gasoline vapor . . a = 47.14 cubic feet.
Add 15 per cent excess air . . . . 7.07 cubic feet.
Total volume of air to be supplied per cubic foot of gasoline vapor . . . xa = 54.21 cubic feet.
The volume of the expanded normal charge containing one cubic foot of gasoline vapor

$$\frac{V_a}{V_o}(xa + 1) = 1.15 \times 55.21 = 63.49 \text{ cubic feet.}$$

Heating-value per cubic foot of gasoline vapor

$$\frac{18,500}{4.2} = 4,400 \text{ B.T.U.}$$

Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(xa + 1)} = \frac{4,400}{63.49} = 69.3 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P., per minute, assuming the compression ratio to be 4, and hence the expected efficiency not less than 0.24.

$$D_1 = \frac{42.42}{Efy\frac{H}{\frac{V_a}{V_o}(xa + 1)}} = \frac{42.42}{0.24 \times 69.3},$$

$$\therefore D_1 = 2.55 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P. per minute, allowing 15 per cent. minimum overload capacity,

$$D_2 = 1.15 \frac{42.42}{E f y \frac{H}{\frac{V_a}{V_o}(x a + 1)}} = 1.15 \frac{42.42}{0.24 + 69.3} = 2.93 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P., assuming the mechanical efficiency to be 0.85.

$$D_4 = 1.35 \frac{42.42}{0.24 \times 69.3} = 3.44 \text{ cubic feet per minute.}$$

The capacity of a gasoline-engine of given dimensions will be in B.H.P.

$$\text{B.H.P.} = \frac{l a N}{3.44 \times 3{,}456} = \frac{l a N}{11{,}900}, \text{ approximately.}$$

l being the length of stroke in inches, a the area of the piston in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-volume is based will be found, in Table XI, to approximate 90 pounds per square inch.

The same result is approximated by the formula,

$$m.e.p. = 5.4 \times 0.24 \times 69.3 = 89.81.$$

Kerosene.—Kerosene, like gasoline, is a fuel obtained in the distillation of crude mineral oil. It is a heavier oil than gasoline; its specific gravity varying from 0.79 to 0.82, or from 47° to 41° Baumé at 62° F.

The composition of kerosene is quite closely represented by the formula $C_{10}H_{22}$, the carbon and hydrogen percentages of which are:

Carbon	0.845
Hydrogen	0.155
	1.000

Average samples of the commercial fuel-oil often analyze, approximately,

Carbon	0.845
Hydrogen	0.139
Nitrogen and Oxygen	0.016
	1.000

and they give at calorimeter tests a calorific value of 20,000 to 23,000 B.T.U. per pound.

The distillates from petroleum possess two properties that vary with their specific gravities, and which determine, largely, the arrangements by which they can be used as motor fuels. These are, the flashing-point and the fire-test or burning-point. The former is its volatility or the temperature at which the fuel gives off, when slowly heated, an ignitible vapor, and the latter is the lowest temperature at which the fuel can continue to burn. The flashing-point of kerosene is at 115° F., or thereabout, and the fire-test is at approximately 140° F.

Kerosene, being less readily vaporized and having a fire-test much higher than gasoline, is a safer fuel in respect to fire-risk than gasoline, but at the same time it is somewhat less suitable for use in the gas-engine, because it requires to be heated in order to become vaporized at atmospheric pressure. It is evident that good combustion in the cylinder can be obtained only when the fuel is thoroughly vaporized at the time explosion takes place. It has been found that if the charge, at its final mixture with air or at the admission to the cylinder, becomes cooled to a temperature essentially below the flashing-point of the fuel, then the best results will not be realized at the combustion.

If the lowest temperature of the charge, allowable at its admission, be assumed to be 80° F., the temperature of the final charge at completed suction-stroke becomes approximately 160° F., and thus, at a drop in pressure of 1½ pound, the volume of the final charge becomes

$$V_a = \frac{14.7}{13.2} \cdot \frac{620}{522} V_o = 1.3\,V_o.$$

To obtain this result the fuel is often heated at its carburation

to a temperature of from 300° to 600° F., depending on the amount of air used at the carburation and on the temperature of the air later admixed at the engine.

Vaporized Kerosene Fuel.—At the combustion of kerosene there is formed 1.4 pound of water per pound of fuel, which absorbs for its vaporization 1,500 heat-units from the heat of combustion. Hence, counting the higher calorific value of the fuel as 20,000 B.T.U., the lower value becomes 18,500 B.T.U.

One pound of kerosene-vapor occupies at atmospheric pressure and at 62° F., a volume of approximately * 2.5 cubic feet, and there will be required for its combustion 190 cubic feet of air.

The estimate of the heating-value of a suitable mixture, per cubic foot, will therefore appear as follows:

The volume of air required by analysis,
per cubic foot of kerosene vapor . . $a = 76$ cubic feet.
Add 15 per cent excess air . . . . 11.4 cubic feet.
Total volume of air to be supplied, per
cubic foot of vapor . . . . . . $x\,a = 87.4$ cubic feet.
The volume of the expanded normal charge containing one cubic foot of vapor

$$\frac{V_a}{V_o}(x\,a + 1) = 1.3 \times 88.4 = 114.9 \text{ cubic feet.}$$

Heating-value of kerosene per cubic foot of vapor

$$H = \frac{18{,}500}{2.5} = 7{,}400 \text{ B.T.U.}$$

Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = \frac{7400}{114.9} = 64.4 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P., per minute, assuming the compression ratio to be 4, and, hence, the expected efficiency not less than 0.24.

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* If the composition of the fuel is that expressed by the formula $C_{10}H_{22}$ then the density of the gas, according to equation 95, becomes 4.92 compared with that of air, and its volume, per pound, at 62° F. 2.64 cubic feet.

$$D_1 = \frac{42.42}{E f y \frac{H}{\frac{V_a}{V_o}(x a + 1)}} = \frac{42.42}{0.24 \times 64.4}$$

$$D_1 = 2.74 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P., per minute, allowing 15 per cent minimum overload capacity,

$$D_2 = 1.15 \frac{42.42}{0.24 \times 64.4} = 3.15 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P., per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35 \frac{42.42}{0.24 \times 64.4} = 3.70 \text{ cubic feet per minute.}$$

The capacity of a kerosene-engine of given dimensions will be in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a N}{3.70 \times 3{,}456} = \frac{l\,a N}{12{,}800} \text{ approximately,}$$

l being the length of stroke, in inches; a the area of the piston, in square inches, and N the number of revolutions per minute.

By looking up in Table XI, page 120, the figure nearest to 2.74 for the maximum I.H.P., it will be noticed that this suction displacement is based on a corresponding mean effective pressure of approximately 84 pounds.

The same result is approximated by the formula

$$m.e.p. = 5.4 \times 0.24 \times 64.4 = 83.46 \text{ pounds.}$$

Properties of the Common Fuel-Gases.—Most of the fuel-gases commonly employed for motive purposes vary quite materially in composition and heating-value at different localities, and some, as for instance producer gas, even from time to time, depending on how the generator is manipulated. Any definite analysis for any particular kind of gas cannot therefore be given, or depended on. Each kind of fuel-gas has, however, its own characteristics by which each may generally be distinguished from another. In the following Table XIII, are given some sample compositions and

TABLE XIII.

Properties of Common Fuel Gases.

Name of Gas.	Constituents of the Gas, in Per Cent. Volume.							Calorific Value per Cub. Ft. at 62° F. B. T. U.		Cub. Ft. of Air Req'd. for Combustion, per c. f. of Gas, Theoretically.	Calorific Value per Cub. Ft. of Normal Charge in B. T. U. $\frac{H}{\frac{Va}{Vo}(xa+1)}$
	H.	CO.	CH_4.	C_2H_4.	O.	CO_2.	N.	High.	Low. (H)		
Blast-Furnace Gas.											
English	1.5	27.5	......	......	......	6.0	65.0	95	94	0.70	42.4
Scotch	2.3	24.7	0.8	......	......	5.8	66.4	96	94	0.72	42.0
Swedish (Charcoal)	5.4	24.0	......	......	......	12.6	58.0	95	92	0.71	41.3
United States	5.2	26.8	1.6	......	0.2	8.2	58.0	120	114	0.92	45.0
Coke-Oven Gas.	53.0	8.2	34.5	......	......	2.0	2.3	550	487	4.78	61.1
at Silesia, Germany	44.6	10.9	19.5	2.2	0.3	5.0	17.5	414	370	3.39	61.3
Illuminating Gas.											
London	47.9	3.7	38.6	4.5	0.3	......	5.0	633	562	5.59	61.5
Glasgow (cannel)	39.2	7.1	40.2	10.0	0.7	0.3	2.4	719	646	6.40	62.8
Paris	52.8	5.6	32.3	5.5	0.5	......	3.8	607	540	5.28	62.1
Carbureted Water-Gas.											
New York	30.3	26.5	24.3	15.0	0.5	1.0	2.4	674	617	5.84	65.0
Worcester	37.2	28.6	18.9	12.9	0.1	0.1	2.2	615	562	5.23	65.1
Lowe Process, San Francisco California Light and Fuel Co.	48.0	7.0	28.0	10.	0.2	2.0	4.8	624	561	5.39	63.4

Natural Gas.											
Anderson, Ind	2.01	0.73	93.07	0.47	0.42	0.26	3.02	960	862	9.06	62
Kokomo, Ind.	1.7	0.55	94.16	0.30	0.30	0.29	2.80	966	868	9.13	62
Marion, Ind.	1.4	0.60	93.57	0.15	0.55	0.30	3.42	957	860	9.04	62
Muncie, Ind.	2.5	0.4	92.67	0.25	0.35	......	3.53	954	858	9.00	61.3
Findley, Ohio	1.84	0.41	93.35	0.35	0.39	......	3.41	959	861	9.06	62
St. Mary's, Ohio	2.14	0.44	93.85	0.20	0.35	......	2.98	963	865	9.10	62
Pittsburg, Pa.	20.0	1.00	72.18	6.30	0.8	0.8	0.00	902	810	8.33	62
Producer Gas Anthracite (Suct.)											
Fair	8.2	22.0	2.4	......	......	6.4	61.0	122	116	0.96	45
Rich	12.0	27.0	1.4	......	......	2.5	57.1	141	134	1.10	48
Average	9.4	23.6	2.6	......	......	5.3	59.1	133	126	1.06	46
Producer Gas, Bituminous.											
Rotative ash table Type	13.8	20.4	3.4	......	......	9.2	53.2	145	134	1.15	50
Rotative ash table Type	12.4	19.2	3.1	......	......	9.5	55.8	134	124	1.06	45
(Mean of 54 Tests on various coals)											
Same producer (Average of 4 Tests on Lignite)	14.6	22.6	3.0	......	......	9.1	50.7	150	140	1.18	48
Water-bottom Type	15.3	21.5	3.6	......	......	8.5	51.0	156	144	1.23	49
Mond Gas	29.0	12.0	2.0	......	......	14.5	42.5	154	137	1.18	47
Water-Gas uncarbureted	51.8	43.4	......	......	......	3.5	1.3	310	282	2.28	63
" " "	49.5	36.0	1.0	......	......	4.2	9.3	289	262	2.06	63
" " "	30.0	28.0	......	......	......	34.0	8.0	189	174	1.39	55

heating-values of various gases that have been duly analyzed and reported on.

Natural Gas as Fuel.—Natural gas varies to some extent in respect to its calorific value, and, to be on the safe side in estimating the cylinder-capacity required for a specified number of horse-power, its heating-value should be appraised conservatively. Its low value may be assumed to be 860 B.T.U. per cubic foot of gas of standard temperature and pressure.

According to analysis there will be required for its combustion, on an average, 9.0 cubic feet of air per cubic foot of gas.

The estimate for the heating-value per cubic foot of suitable mixture becomes then:

The volume of air required by analysis per cubic foot of gas . . . . . . $a =$ 9.0 cubic feet.
Add 15 per cent excess air . . . 1.35 cubic feet.
Total volume of air to be supplied, per cubic foot of gas . . . . $x\,a =$ 10.35 cubic feet.
Total volume of the mixture, per cubic foot of gas . . . . . . $x\,a + 1 =$ 11.35 cubic feet.
The volume of the expanded normal charge containing one cubic foot of gas

$$\frac{V_a}{V_o}(x\,a + 1) = 1.23 \times 11.35 = 13.96 \text{ cubic feet.}$$

Heating-value per cubic foot of gas . . . $H =$ 860 B.T.U.
Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = \frac{860}{13.96} = 62 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P., per minute, assuming the compression-ratio to be 5, and, hence, the expected efficiency not less than 0.26,

$$D_1 = \frac{42.42}{E\,f\,y\,\frac{H}{\frac{V_a}{V_o}(x\,a + 1)}} = \frac{42.42}{0.26 \times 62}$$

$$D_1 = 2.63 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P., per minute, allowing 15 per cent minimum overload capacity,

$$D_2 = 1.15 \frac{42.42}{0.26 \times 62} = 3.02 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P., per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35 \frac{42.42}{0.26 \times 62} = 3.55 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions when running on natural gas will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,aN}{3.55 \times 3{,}456} = \frac{l\,aN}{12{,}300},$$

l being the length of stroke, in inches, a the area of the piston, in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based is approximately 88 pounds. See Table XI, page 120.

Illuminating-Gas.—City illuminating-gas is still, occasionally, and it was not many years ago, generally, obtained by distilling off the volatile hydrocarbons from bituminous coal. The condensible products are, partly, fixed by being heated to a high temperature in the retorts, and, partly, removed at the cooling and cleaning process. The result of the process is a gas consisting mainly of hydrogen, methane and heavy hydrocarbons (illuminants). A quite average sample of coal-gas is the analysis, given by M. W. Robinson, of the Birmingham, England, city-gas, the mean composition of which approximates, by volume,

	H	CH_4	C_2H_4	CO	CO_2	air.
per cent	45	40	5	5	1	4.

At 62° Fahrenheit, the volume per pound of gas is 32 cubic feet.

The heating-value of the gas is 650 B.T.U. per cubic foot, and the volume of air required for its combustion is 5.75 cubic feet per cubic foot of gas.

In the samples of illuminating-gas given in Table XIII, the heating-value varies from 607 to 719 B.T.U. per cubic foot, and

the calorific value per cubic foot of expanded normal charge, including 15 per cent excess air, is from 61.5 to 62.8. With these figures as basis for a computation the required displacement volume of the cylinder, or the normal mean effective pressure, can readily be obtained, identically with the computation for natural gas.

Illuminating-gas is at present often produced by adding illuminants to so-called water-gas of a proper heating-value to imitate the old-fashioned retorted coal-gas. The fuel-value in this manufactured gas consists principally of hydrogen and carbon monoxide, but its actual composition varies quite materially, depending on the manipulation of the water-gas generators. The gas is somewhat lighter than coal-gas, and its heating-value varies between 550 to 600 B.T.U. per cubic foot. The air required for its combustion in the gas-engine is the same as that required by coal-gas, or between 5 to 6 cubic feet per cubic foot of gas.

Coke-Oven Gas.—Modern coke-oven plants yield considerable quantities of gas, in excess of the requirements for fuel for the coking-process. This gas is becoming more and more to be considered an important fuel for generating power in the gas-engine. The yield of gas is, however, a fluctuating factor; as it varies with the quality of the fuel, and with the different stages of the coking-process. The output of the coke-ovens, moreover, varies with the fluctuating demands from the furnaces that absorb their product, and this fact must be taken into consideration in estimating the actual value of the gas for independent industrial use.

The more recent types of regenerative coke-ovens yield, per net ton of coal, on an average 6,000 to 10,000 cubic feet of gas, respectively from coals low and from those high in volatile matter. Of this, about 40 per cent will be available as a by-product for industrial purposes—and the heating-value of the gas will vary from 500 to 700 B.T.U. per cubic foot.

The average total yield of gas may be assumed 8,000 cubic feet of a heating-value of 600 B.T.U., or 4,800,000 B.T.U. per net ton of coal. As the coking process proceeds, practically, during 24 hours, there will be obtained 200,000 B.T.U. per hour;

40 per cent of which is 80,000 B.T.U., available, on an average per hour, for industrial purposes. Utilized in an engine of an economy of, say, 10,000 B.T.U. per indicated horse-power per hour, there would, thus, be generated 8 * indicated horse-power per net ton of coal coked, which in a plant of the moderate capacity of 500 tons per day would amount to a power-reserve of 4,000 indicated horse-power, continuously.

There being connected with economical plants a process for the recovery of the by-products in the tar which is collected at the cooling of the gas, the gas will be supplied from the coking plant already partially cleaned. The tar precipitators or separators generally employed are simply high steel-plate cylinders through which the gas passes slowly up or down in a zig-zag path, baffled by shelvings, to the right and to the left, on which the tar is deposited as the gas is gradually cooled.

For a further cleaning of the gas, to make it suitable for use in the gas-engine, centrifugal cleaners, similar to those used for the cleaning of bituminous producer-gas, may be employed. The final cleaning is accomplished in so-called dry scrubbers, which are identical to those described in connection with producer-gas installations, page 436.

The composition and heating-value of the gas as well as the supply of air required for its combustion are factors that vary somewhat. For the determination of the required suction-displacement per horse-power of the engine using this gas, it may be assumed that the lower heating-value of the gas is not less than 500 B.T.U. per cubic foot and that the air required for its proper combustion, including 15 per cent excess above the theoretical requirement, is 8 cubic feet per cubic foot of gas. The heating-value of the expanded normal charge will then be 60 B.T.U. per cubic foot, or nearly at par with that of illuminating gas.

Bituminous Producer-Gas as Fuel.—The low calorific value of bituminous gas should, on an average, not be less than 138 B.T.U per cubic foot at standard temperature and pressure, and for its combustion there will be required, according to analysis, a mean

* 10 to 12 horse-power is sometimes, less conservatively, stated.

of 1.2 cubic foot of air per cubic foot of gas. In order to obtain a complete combustion in the gas-engine cylinder it is, however, required that there should be an excess of air of in the neighborhood of 15 per cent.

Founded on these data, the estimate for the heating-value per cubic foot of suitable mixture and for the required cylinder capacity becomes:

The volume of air required by analysis per cubic foot of gas . . . . . $a = 1.2$ cubic foot.
Add 15 per cent excess air . . . . 0.18 cubic foot.

The total volume of air to be supplied per cubic foot of gas . . . . . . $xa = 1.38$ cubic foot.

The volume of the mixture per cubic foot of gas, at standard temperature and pressure . . . . $xa + 1 = 2.38$ cubic feet.

The volume of the expanded normal charge containing one cubic foot of gas

$$\frac{V_a}{V_o}(xa + 1) = 1.23 \times 2.38 = 2.93 \text{ cubic feet.}$$

Heating-value per cubic foot of standard gas $H = 138$ B.T.U.
Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(xa + 1)} = \frac{138}{2.93} = 47 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P. per minute, assuming the compression ratio to be 6.5, and the expected efficiency, thus, not less than 0.3,

$$D_1 \quad \frac{42.42}{Efy\frac{H}{\frac{V_a}{V_o}(xa \times 1)}} - \frac{42.42}{0.3 \times 47}$$

$$D_1 = 3.01 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P. per minute, allowing 15 per cent minimum overload capacity,

$$D_2 = 1.15\frac{42.42}{0.3 \times 47} = 3.46 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P. per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35 \frac{42.42}{0.3 \times 4.7} = 4.06 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions, when running on bituminous producer-gas, will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a\,N}{4.06 \times 3{,}456} = \frac{l\,a\,N}{14{,}000}$$

l being the length of stroke, in inches, a the area of the piston, in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based will be found in Table XI, page 120, opposite the values $D_1 = 3.02$, $D_2 = 3.47$ and $D_4 = 4.08$. Hence, M.E.P. = 76 pounds per square inch. The same value can be computed from the relation

$$m.e.p. = 5.4\, E\, f\, y \frac{H}{\frac{V_a}{V_o}(x\,a + 1)}$$

Anthracite Suction Producer-Gas as Fuel.—Anthracite suction gas of efficient quality has, on an average, a low calorific value of not less than 120 B.T.U. per cubic foot, and for its combustion there is required, according to analysis, 1.1 cubic foot of air per cubic foot of gas. An excess of air of about 15 per cent should, however, be provided, in order to insure complete combustion in the gas-engine cylinder.

The estimate for the heating-value per cubic foot of suitable mixture becomes:

The volume of air required by analysis per cubic foot of gas	$a = 1.1$	cubic foot.
Add 15 per cent excess air	0.165	cubic foot.
The total volume of air to be supplied per cubic foot of gas	$x\,a = 1.265$	cubic feet.
The volume of the mixture, per cubic foot at standard temperature and pressure	$x\,a + 1 = 2.265$	cubic feet.

The volume of the expanded normal charge containing one cubic foot of gas

$$\frac{V_a}{V_o}(x\,a + 1) = 1.23 \times 2.265 = 2.78 \text{ cubic feet.}$$

Heating-value per cubic foot of standard gas $H = 120$ B.T.U.
Heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = \frac{120}{2.78} = 43.2 \text{ B.T.U.}$$

The minimum suction-displacement necessary per I.H.P. per minute, assuming the compression ratio to be 6.5 and the expected efficiency, thus, not less than 0.3:

$$D_1 = \frac{42.42}{E f y \frac{H}{\frac{V_a}{V_o}(x\,a + 1)}} = \frac{42.42}{0.3 \times 43.2}$$

$$D_1 = 3.27 \text{ cubic feet per minute.}$$

The required suction-displacement per rated I.H.P., per minute, allowing 15 per cent minimum overload capacity:

$$D_2 = 1.15 \frac{42.42}{0.3 \times 43.2} = 3.76 \text{ cubic feet per minute.}$$

The required suction-displacement per rated B.H.P. per minute, assuming the mechanical efficiency to be 0.85,

$$D_4 = 1.35 \frac{42.42}{0.3 \times 43.2} = 4.41 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions when running on suction gas will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a\,N}{4.41 \times 34{,}56} = \frac{l\,a\,N}{15{,}240}, \text{ approximately.}$$

l being the length of stroke, in inches, a the area of the piston, in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based will be found in Table XI, page 120. It

approximates the value opposite $D = 3.27$ or 70 pounds per square inch.

The same result will be approximated by solving equation 50*b*.

Thus, $m.e.p. = 5.4 \times 0.3 \times 43.2 = 69.98$ pounds.

Comparing the figures for the suction-displacement and the mean effective pressure obtained for anthracite suction gas with those obtained for bituminous gas, it will be seen, that they vary in the same ratio as the heating-value per cubic foot of normal expanded charge. That this must be so is evident, because, other things being equal, in the rate that more heat-units are put into an engine for transformation in that rate its capacity for doing work must increase.

The Engine-Power at an Elevation above the Sea-Level. The normal suction-displacement per horse-power and the mean effective pressure that may be derived from the various gas-engine fuels has in the preceding paragraphs been determined on the basis that the charge is supplied to the engine under the pressure of the atmosphere at the sea-level (14.7 pounds). The fact that the gas alone, perhaps, is supplied at a pressure somewhat higher, or lower, than the atmosphere will not materially affect the power of an engine, as long as the air for the charge is supplied under the pressure of the free atmosphere.

If an engine were to work under an atmospheric pressure less than 14.7 pounds, as would be the case if installed at a high altitude, then this fact must be taken into consideration in determining its power.

The heating-value of the expanded normal charge at sea-level is

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} \quad \text{or} \quad \frac{H}{\frac{p_o}{p_a}\frac{T_a}{T_o}(x\,a + 1)}.$$

For an atmospheric pressure 14.7 pounds, p_a has been assumed to be ($p_o - 1\frac{1}{2}$) 13.2 pounds, or a total drop in pressure between the outside and the pressure in the cylinder at the end of the suction-stroke of $1\frac{1}{2}$ pound has been allowed. This normal drop in pressure would presumably become proportionately less as the atmosphere becomes rarefied at altitudes elevated above the sea-level.

Thus, the pressure in the cylinder at the higher altitude

$$p_a' = \frac{p_o'}{p_o}(p_o - 1\tfrac{1}{2}),$$

when p_o' is the local atmospheric pressure, in pounds.

Or,
$$p_a' = \frac{B_o'}{B_o}(p_o - 1\tfrac{1}{2}),$$

when B_o' is the local barometric pressure and B_o the barometric pressure at the sea-level. The heating-value of the expanded normal charge under the barometric pressure B_o' becomes

$$\frac{B_o'}{B_o} \frac{H}{\frac{p}{p_a}\frac{T_a}{T_o}(x\,a + 1)},$$

or it becomes reduced in the ratio $\frac{B_o'}{B_o}$.

As the heating-value of the expanded normal charge determines the mean effective pressure, and the horse-power of an engine, therefore these quantities will, under a barometric pressure B_o', also be reduced in the ratio $\frac{B_o'}{B_o}$.

The horse-power, $H.P._{el}$, at an elevation compared with the horse-power at the sea-level, $H.P_{sl}$, becomes, thus

$$H.P._{el} = \frac{B_o'}{B_o} H.P._{sl}.$$

Example.—What will be the normal rated brake horse-power of a 16 × 24 four-cycle engine, 200 revolutions per minute, working on anthracite producer-gas at an altitude of 4,000 feet?

The normal barometric pressure at an elevation of 4,000 feet is 26″ inches mercury.

thus,
$$H.P._{4000} = \frac{26}{30} H.P._{sl}.$$

The 16 × 24 engine at the sea-level is, according to Table XXV, page 274, of 62 rated brake horse-power. Hence at an elevation of 4,000 feet it is $\frac{26}{30} \times 62 = 0.87 \times 62 = 54$ rated brake horse-power.

Blast-Furnace Gas.—The, so-called, waste gas from blast-furnaces has for years been an important fuel in connection with the operation of the furnaces. Of greatest importance is its use for the heating of the hot-blast stoves and for generating steam by which blowing-engines and dynamos are operated. For the latter purpose, however, the gas has proven not very efficient, and is at times even too lean to burn satisfactorily. Of late years, the gas has been extensively used in large gas-engines, for generating abundantly the motive power required for blast-engines and auxiliaries and by this system of utilizing the gas there has been accomplished a material saving in expense for additional fuel.

For the production of one ton of pig-iron there is used, on an average, one ton of coke; and, approximately, six tons* of gas are generated. Estimates place the requirement of gas for the hot-blast stoves, and other heating purposes, at about 30 per cent of the total output, or at about 3,600 pounds per ton of pig; the remaining 8,400 pounds of gas will, therefore, be available for power purposes.

The analysis of an average sample of the gas would show it to be of a heating-value of approximately 1,320 B.T.U. per pound, or a total of 11,088,000 B.T.U. in 8,400 pounds. Assuming an engine utilizing this gas to have a thermal efficiency of 0.24, which is common with respect to large engines, there would, then in the way of power, be obtained, per ton of pig-iron produced per hour, $\frac{11{,}088{,}000 \times 0.24}{2{,}545} = 1{,}040$ indicated horse-power, or approximately 884 † brake horse-power.

The power required for driving blowing-engines, pumps, etc., may be liberally figured at 284 horse-power per ton of pig-iron produced per hour; hence there will remain approximately 600 horse-power, in excess of the requirements for the furnace.

The gas is, after leaving the top of the blast-furnace, carried through the down-comer into a system of dust-catchers, which

* Corresponding to 158,000 cubic feet per ton of pig. The volume is variously estimated at from 180,000 to 120,000 cubic feet, varying with the class of ore and with the amount of coal consumed.

† Variously estimated at from 750 to 1,200 B.H.P.

collect the greater part of the dust with which the gas is charged, and it is in this manner made clean enough for use as fuel. For being utilized in the gas-engine it is, however, required that the gas be further cleaned, to eliminate practically all the fine dust, which, otherwise, will prove injurious to the engine.

Fig. 36 is a diagrammatic view of a washing-plant for blast-furnace gas.

After having deposited a large part of its dust in the dust-collector, the gas is led into the tower washer, which consists of three high cylinders containing, each, several grids over which a shower of water is sprayed from spray pipes above. The gas is passed up through one after the other of these cylinders, as plainly indicated by the small arrows in the figure, and when leaving the apparatus its dust-contents has become reduced to 1.5 to 1.0 gramme per cubic metre * of gas.

From the primary washer the gas is passed into the centrifugal cleaner, which in the figure is a Thiesen washer.

This apparatus consists of a casing containing a revolving drum which is provided with helical vanes on its outside surface. Between the drum, which revolves at a speed of 350 revolutions per minute, and the casing the gas is passed and is vigorously thrown out against a film of water passing over the inside surface of the casing; the water entering through valves at *a* and drained off, together with the dust contents, into a drain and seal basin at *b*.

The gas is drawn in to the washer by means of a fan driven by the drum shaft and located in the casing at *c*; the gas-current is retarded by the helical drum-vanes, and then again given a suitable discharge-speed by a fan located at the exit end of the casing, at *d*.

An apparatus of this type, cleaning 720,000 to 1,000,000 cubic feet of gas per hour, requires a motor of about 150 horse-power; the power-requirements being figured generally 0.15 to 0.2 horse-power per 1,000 cubic feet of gas. The gas leaving the washer is customarily tested for dust, at intervals, and the quantity runs normally from 0.01 to 0.02 gramme per cubic metre of gas.

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* Equivalent, practically, to 0.66 to 0.44 grains per cubic foot.

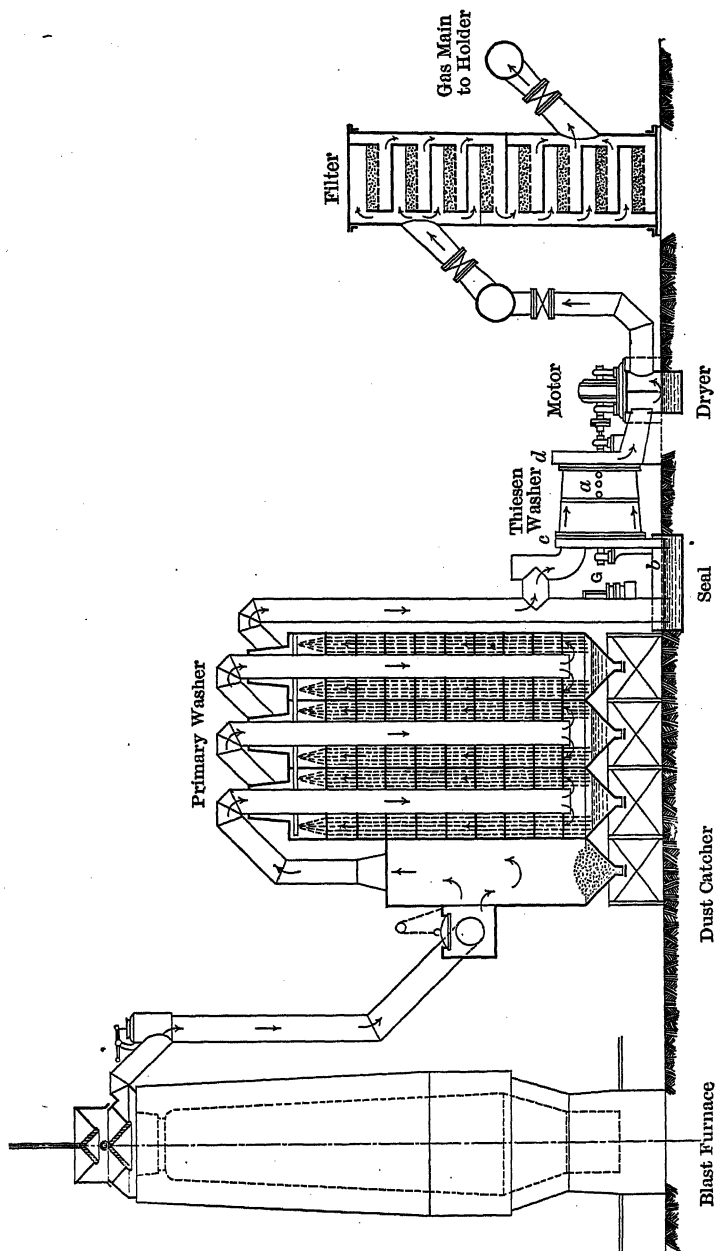



FIG. 36.—Washing Plant for Blast-Furnace Gas.

From the washer the gas is conducted, through a dryer, to a filtering-tower, which consists of a cylinder containing a double set of shelves charged with filtering material through which the gas passes, in the manner plainly indicated by the small arrows in the figure, and from there it is delivered to the gas-holder and the engine.

At *G* is shown a gasometer, which has for purpose to shut down the Thiesen washer the moment the pressure in the delivery pipe from the primary washer becomes dangerously close to that of the atmosphere. Without such an apparatus the plant would be unsafe, as, with the washer continuing to draw from the primary washer after the gas might have been shut off from the furnace, air would be drawn in to the gas-system, at the risk of producing an explosive mixture in the holder and gas-main.

The gas used in the hot-blast stoves and boilers is, according to the latest practice, passed from the dust-catcher also through a simple stationary washer, to prevent the choking of gas-flues by dust, which otherwise may occur.

The composition of blast-furnace gas varies considerably, but it contains, under normal conditions, only a small percentage of hydrogen, which makes it suitable for a high compression in the gas-engine. At times, however, a very large percentage of hydrogen may be present, and this fact must be taken account of, when deciding upon the proper compression to be carried by the engine. While a compression-pressure of 190 pounds has been frequently used for this gas, it has, due to difficulties, recently been cut down considerably. A pressure of approximately 160 pounds should be considered safe.

The following is a normal composition of blast-furnace gas:

	Per Cent Volume	Per Cent Weight
CO	27.0	26.32
H	2.5	0.18
CH_4	0.5	0.29
CO_2	9.0	13.79
N	61.0	59.42
	100.0	100.00

Heating-value per cubic foot at 62° F., and 1 atmosphere 102 B.T.U.

Volume per pound of gas at 62° F. and 1 atmosphere 13.2 cubic feet.

For complete combustion of one pound of gas there is required 0.76 pound of air; or, generally, there is required per cubic foot of gas approximately, ¾ cubic foot of air.

Including 15 per cent excess air, there should be supplied for combustion in the gas-engine 0.86 cubic foot of air per cubic foot of gas.

CHAPTER VII

ALCOHOL FUELS

In France and Germany there has for a number of years been made a considerable effort to encourage the use of alcohol as fuel for motor purposes, and to replace, as much as possible, gasoline (an imported commodity) by alcohol. In the United States there has also developed, lately, a purpose to extend the markets for fuel-supply beyond the control of a mining-monopoly, and to open them for the modern fuel, in the interests of the agriculture of the nation.

The ample investigations made for the solution of the proposition make it appear certain, at present, that, as an engineering proposition, the problem is solved, and it remains an open question only so far as its proper economy is concerned.

The Alcohols.—The alcohols manufactured are of two kinds: ethyl alcohol, C_2H_6O, obtained through fermentation and distillation of sugar or starch; and methyl alcohol, CH_4O, obtained by destructive distillation of wood.

The following table contains some of the principal data regarding these products in their pure state.

Product.	Specific Gravity.		Specific Heat.		Boiling Point at 760 m/m Degrees Fahr.	Latent Heat, in Vapor.		Vapor Tension. Lbs. per sq. in.
	At 32° F.	At 62° F.	Of Gas at const. press.	Of Liquid.		From 62° F.	Above Boiling Point.	
Ethyl Alcohol C_2H_6O	0.806	0.794	0.453	0.726	173	440	365	0.7
Methyl Alcohol CH_4O	0.810	0.798	0.458	0.670	148	510	475	1.5

For the complete combustion of the alcohols there is required:

	Oxygen per pound of fuel	Air per pound of fuel
by C_2H_6O	2.086 pounds	9.03 pounds
by CH_4O	1.5 pounds	6.5 pounds

Compared with gasoline, the alcohols are of rather low heating-value; when pure, their calorific power compares with that of the former fuel as follows:

Gasoline	20,000 B.T.U.
Ethyl alcohol	13,000 B.T.U.
Methyl alcohol	10,000 B.T.U.

The manufactured alcohols are never pure, however, but hydrated to the extent of containing from 7 to 50 per cent water, and their actual heating-values, as fuels, will be on an average only:

for Ethyl alcohol (water contents 14%) 13,700 B.T.U.
for Methyl alcohol (water contents 14%) 11,000 B.T.U.

One requirement, in order to make it practical to allow the free production and sale of alcohol fuels on a large scale, was to find a suitable denaturant that would make the product unusable for other purposes than as fuel. There prevail, for the purpose, two satisfactory methods, the results of which are generally termed, respectively, "denatured alcohol" and "carbureted alcohol."

The composition of the former mixture is defined by law in the United States and in France; it being required that: "To 100 volumes ethyl or grain alcohol of a strength of not less than 90 per cent there must be added 10 volumes methyl or wood alcohol and ½ volume of a heavy hydrocarbon, pyridine—denaturation benzol."

The latter ingredient, required to have the boiling-points between the temperatures 280° and 390° F., is added for the purpose only of giving to the fuel a distinctive color and odor that can readily be recognized at inspection.

Carbureted alcohol is a mixture of any optional proportion

of the denatured alcohol with benzol; the latter being a purer article than that used for denaturing purposes.

Benzol and pyridine are products from by-product coke-ovens. They are increasingly used abroad for the enriching and denaturing of alcohol for motor purposes, and as illuminants in manufactured gas. Outside of their uses for these purposes, they are valuable raw materials in manufacturing chemicals. Commercial 90 degrees benzol, of a specific gravity of 0.88 at 62° F. and boiling between the temperatures 200° and 300° F., is of about 18,000 B.T.U. per pound, or of a heating-value a little less than that of gasoline.

A mixture of equal parts of denatured alcohol and benzol has been extensively tried for motor use, and in France a series of competitive tests have been made with this fuel and with pure denatured alcohol to ascertain their relative advantages *

The analyses of the two fuels were as follows:

	Denatured Alcohol	Carbureted Alcohol 50 per cent Benzol
Carbon, C	0.4372	0.6899
Hydrogen, H	0.1112	0.0948
Oxygen, O	0.3029	0.1457
Water, H_2O	0.1408	0.0685
	0.9921	0.9989
Computed higher heating-value	9,938 B.T.U.	14,225 B.T.U.
Heating-value by test	10,630 B.T.U.	14,180 B.T.U.

Some of the results obtained at the tests were:

Fuel Consumption; Ratio by Weight

Power Developed B.H.P.	Denatured Alcohol	50% Carbureted Alcohol
8.3	10	7.66
16.3	10	6.85
34.4	10	7.61

The conclusions were, that, for the same power, the consumption of 50 per cent carbureted alcohol will average $\frac{7}{10}$ of that of denatured alcohol.

* Concours International De Moteurs et Appareils utilisant L'Alcool dénaturé 1902.

Numerous competitive tests have also been made between denatured alcohol and gasoline, the results of which tend to show that, on an average, the consumption of denatured alcohol will be 1.05 pound per brake horse-power per hour against a consumption of 0.7 pound of gasoline, or, for equal power, in the ratio of 1½ pound of alcohol to 1 pound of gasoline. The latter ratio expressed in volumes would be 1.38 gallon of alcohol to 1 gallon of gasoline. Counting the lower heating-value of gasoline 18,500 and that of denatured alcohol 10,000 B. T. U., the efficiency of the two fuels becomes in the ratio of 24 for gasoline to 30 for alcohol.

Comparisons between the two fuels have been stated in different ways, and the figures will vary, to some extent, with the type of engine to which they have reference. For instance, during the sessions of the Motor Union of Great Britain and Ireland, which had for purpose to investigate the relative values, as motor-fuels, of gasoline and alcohol, it was stated: "It has been brought out through evidence that petrol (gasoline), and alcohol stand in the ratio of 2 to 1 as regards their heat of combustion, but that in the case of alcohol 30 per cent of the heat is available, while in the case of petrol 20 per cent can be obtained. We require then 4 parts of alcohol to 3 of petrol, by weight."

The Elementary Components of Alcohol Fuels.—The specific gravities of the two fuel-alcohols being so nearly the same, and varying somewhat with the purity of the products, any distinction between these quantities will be of no purpose in any ordinary computation. It will be assumed in the following that both are 0.8 at 62° F.

The specific gravity of benzol is 0.88 and that of water at 62° F. may be called 1.0.

Example.—Denatured alcohol consists of:

90 volumes of ethyl alcohol, $C_2 H_6 O$
10 volumes of water, $H_2 O$
10 volumes of methyl alcohol, $C H_4 O$
0.5 volumes of benzol, $C_6 H_6$.

What are the weight percentages of the elementary components?

The following scheme for a dissecting analysis may conveniently be used:

Specific Weight.	Ratio of Volumes of Components.	Ratio of Weights of Components.	Weight percentages of Components.	Symbol for Components.	Mol. Weights of Components.	Ratio Between Weights of Elements.			Percentages of Weights of Elements.		
						C.	H.	O.	C.	H.	O.
1	2	3	4	5	6	7	8	9	10	11	12
0.8	90	72	0.7961	C_2H_6O 24 6 16	46	$\frac{24}{46}$	$\frac{6}{46}$	$\frac{16}{46}$	0.4154	0.1038	0.2770
1.0	10	10	0.1106	H_2O 2 16	18	.....	$\frac{2}{18}$	$\frac{16}{18}$	......	0.0123	0.0984
0.8	10	8	0.0885	CH_4O 12 4 16	32	$\frac{12}{32}$	$\frac{4}{32}$	$\frac{16}{32}$	0.0333	0.0111	0.0444
0.88	0.5	0.44	0.0048	C_6H_6 72 6	78	$\frac{72}{78}$	$\frac{6}{78}$	.....	0.0044	0.00037	......
	110.5	90.44	1.0000						0.4531	0.12757	0.4198

How the figures in the above table have been derived is evident. It may be necessary only to state that the figures of the last three columns are the products of the weights of elements, columns 7, 8 and 9, and corresponding weight percentages of components, column 4.

The percentages of the elements are:

$$C = 0.453$$
$$H = 0.127$$
$$O = 0.420$$

The water contents having no influence in respect to the heating-value of a fuel, it may be desired to keep it separate.

The percentages in that case will be:

$$C = 0.453$$
$$H = 0.115$$
$$O = 0.321$$
$$H_2O = 0.111$$

The specific weight of denatured alcohol will be $\frac{90.44}{110.5} =$ 0.8185 or, approximately, 0.818 at 62° F.

Assuming that a carbureted alcohol-fuel consists of equal

volumes of denatured alcohol and benzol, the weight percentages of the elements of the mixture will be:

Components.	Specific Weight.	Weight percentages of Components.	Ratio of Weights of Elements.				Percentages of Weights of Elements.			
			C.	*H.*	*O.*	H_2O.	*C.*	*H.*	*O.*	H_2O.
Denatured alcohol	0.818	0.4817	0.453	0.115	0.321	0.111	0.21821	0.05540	0.15463	0.05346
Benzol	0.88	0.5183	$\frac{72}{78}$	$\frac{6}{78}$	.....	.....	0.47833	0.04000	....	....
	1.698	1.0000					0.69655	0.09540	0.15463	0.05346

The weight-percentages of the elements, thus,

$$C = 0.697$$
$$H = 0.095$$
$$O = 0.155$$
$$H_2O = 0.053$$

The weight-percentages of the components of 50 per cent carbureted alcohol are:

Components.	Ratio of Volumes.	Specific Weights.	Ratio of Weights.	Percentage Weight.
Denatured alcohol	0.5	0.818	0.409	0.4816
Benzol	0.5	0.88.	0.44	0.5184
			0.849	1.0000

In France and in the United States the strength of alcohol is uniformly expressed by stating what percentage *volume* there is of pure alcohol in a diluted mixture. In Germany, however, its strength is always expressed by the percentage *weight* the pure alcohol is of the whole. In comparing results, therefore, reported from the different countries, this difference should be allowed for. For instance, an alcohol called in this country 90 per cent will be in Germany 0.858 per cent. For the reduction from per cent volume to per cent weight, or vice versa, a table of experimental determinations of the density of different hydrations will be necessary. In the Smithsonian table, below, for ethyl alcohol, the densities are given at 60° F., and it should be observed that

for each degree the temperature of the fluid is higher or lower than 60° its density will be approximately 0.0005 less or more than that given in the table.

Specific Gravity of Ethyl Alcohol Diluted by Various Percentages of Water.

Specific Gravity at 60° F. compared with Water at 60° F.	Percentage of Alcohol. By Weight.	Percentage of Alcohol. By Volume.	Specific Gravity at 60° F. compared with Water at 60° F.	Percentage of Alcohol. By Weight.	Percentage of Alcohol. By Volume.
0.834	85.8	90.0	0.822	90.4	93.4
.833	86.2	90.3	.821	90.8	93.7
.832	86.6	90.6	.820	91.1	94.0
.831	87.0	90.9	.819	91.5	94.2
.830	87.4	91.2	.818	91.9	94.5
.829	87.7	91.5	.817	92.2	94.8
.828	88.1	91.8	.816	92.6	95.0
.827	88.5	92.1	.815	93.0	95.3
.826	88.9	92.3	.814	93.3	95.5
.825	89.3	92.6	.813	93.7	95.8
.824	89.6	92.9	.812	94.0	96.0
.823	90.0	93.2			

In the custom house alcohol is given in "proof" gallons, that is, in gallons containing 50 per cent alcohol, by volume, 50 per cent being water. When a quantity of alcohol is given in proof gallons it is, therefore, expressed by a figure twice as large as it would be if the measure were given in gallons of 100 per cent alcohol.

Specific Heat of the Fuel-Vapors.—Denatured alcohol consists of:

Ethyl alcohol .	C_2H_6O	0.796 per cent weight.
Methyl alcohol	CH_4O	0.088 " " "
Benzol	C_6H_6	0.005 " " "
Water	H_2O	0.111 " " "

50 per cent carbureted alcohol consists of:

Ethyl alcohol	C_2H_6O	0.383 per cent weight.
Methyl alcohol	CH_4O	0.042 " " "
Benzol	C_6H_6	0.521 " " "
Water	H_2O	0.054 " " "
		1.000

Based on these compositions of the two fuel-alcohols we obtain:

The specific heat of denatured alcohol-vapor.

C_2H_6O	$0.796\times 0.453=0.360$
CH_4O	$0.088\times 0.458=0.0403$
C_6H_6	$0.005\times 0.3 \ =0.0015$
H_2O	$0.111\times 0.48 \ =0.0532$
Total	0.455

The specific heat of 50 per cent carbureted alcohol-vapor.

C_2H_6O	$0.383\times 0.453=0.173$
CH_4O	$0.042\times 0.458=0.019$
C_6H_6	$0.521\times 0.3 \ =0.156$
H_2O	$0.054\times 0.48 \ =0.026$
Total	0.374

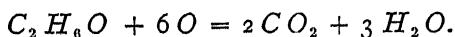
Air Required for Combustion.—In the following table are given the oxygen and air required for the complete combustion of the alcohol-fuels and components, and the combustion-products obtained per pound of fuel:

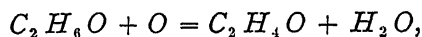
Fuel.	Oxygen Required for Combustion. Lbs.	Air Required for Combustion. Lbs.	Combustion-Products.		
			CO_2. Lbs.	H_2O. Lbs.	N. Lbs.
Ethyl alcohol, C_2H_6O ..	2.087	9.037	1.913	1.174	6.95
Methyl alcohol, CH_4O ..	1.5	6.50	1.375	1.125	5.00
Benzol, C_6H_6	3.1	14.32	3.385	0.692	10.243
Denatured alcohol	1.809	7.833	1.661	1.149	6.024
50 % Carbureted alcohol	2.448	10.600	2.530	0.918	8.160

The higher heating-values of the two alcohol-fuels are:

	Computed Value	Value by Test
Heating-value of denatured alcohol	11220	11880
Heating-value of 50% carbureted alcohol (50% alcohol) . . .	14830	14200
Heating-value of 80% carbureted alcohol (80% alcohol) . . .	12050	

At complete combustion of alcohol there is formed carbon dioxide and water, as:

$$C_2H_6O + 6O = 2CO_2 + 3H_2O.$$

When the combustion is incomplete, however, there is formed aldehyde and water, as:

$$C_2H_6O + O = C_2H_4O + H_2O,$$

or acetic acid and water, as:

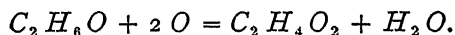
$$C_2H_6O + 2\,O = C_2H_4O_2 + H_2O.$$

Vapor Pressure and Saturation Temperature of an Explosive Gas Mixture.—The following table, according to Sorel, gives the vapor-pressure of saturation at various temperatures for the alcohol-fuels commonly used, or it may be said to give the saturation temperatures for various pressures.

TABLE XIV.

Vapor Pressure of Saturation for Denatured and 50 Per Cent Carbureted Alcohol, in Millimeters of Mercury.

Temperature.		Denatured Alcohol.	Carbureted Alcohol.	Temperature.		Denatured Alcohol.	Carbureted Alcohol.	Temperature.		Denatured Alcohol.	Carbureted Alcohol.
°C.	°F.			°C.	°F.			°C.	°F.		
−14	6.8	....	19.5	0	32	15	43	14	57.2	35	82.5
−13	8.6	....	20.5	1	33.8	16	45	15	59	37	86
−12	10.4	....	22	2	35.6	17	47.5	16	60.8	39.5	90
−11	12.2	....	23	3	37.4	18	50	17	62.6	42	93.5
−10	14	....	24.5	4	39.2	19	52.5	18	64.4	45	97.5
− 9	15.8	....	26	5	41	20	55	19	66.2	48	102
− 8	17.6	....	27.5	6	42.8	21.5	57.5	20	68	51	106.5
− 7	19.4	....	29	7	44.6	23	60.5	21	69.8	54	111.5
− 6	21.2	11.5	31	8	46.4	24.5	63	22	71.6	57.5	117.5
− 5	23	12	32.5	9	48.2	26	66.5	23	73.4	61	123.5
− 4	24.8	12.5	35.	10	50	27.5	70	24	75.2	64	130
− 3	26.6	13	37	11	51.8	29	72.5	25	77	68	136.5
− 2	28.4	13.5	38.5	12	53.6	31	75.5	26	78.8	72	145
− 1	30.2	14.5	41	13	55.4	33	79	27	80.6	76.5	.....

With regard to the alcohol-fuels, it becomes of special interest to find the minimum temperature at which a proper explosive mixture can be formed. This temperature may readily be determined, on the basis of the preceding table, when it is remembered that the ratio of the pressures of, respectively, the air and

the fuel-vapor in a saturated mixture is equal to the direct ratio of the volumes of the two gases reduced to the same pressure.

Let B be the barometric pressure to which a mixture of fuel-vapor and air is subjected,

t and $(a + t)$ its temperature, respectively, above 0° F., and above the absolute zero,

d the density of the vapor at 32°F., and at atmospheric pressure,

ϵ the coefficient of expansion by heat of air and of the vapor,

V the volume of the air mixed with each pound of the fuel, at 32° F., and atmospheric pressure, in cubic feet,

x the vapor-pressure of the fuel,

and, thus, $B - x$ the vapor-pressure of the air in the mixture.

The volume of one pound of vapor, at the temperature t and barometric pressure B, becomes, then, according to equation 11

$$\frac{760}{B}\,\frac{1}{d}\,\epsilon\,(a + t).$$

and the volume of the corresponding proportion of the air is

$$\frac{760}{B}V\,\epsilon\,(a + t).$$

Hence,

$$d\,V$$

$$x = \frac{B}{1 + d\,V}.$$

If the fuel-and-air mixture is subjected to the atmospheric pressure, then the pressure of the vapor becomes

$$x_a \quad \frac{760}{1 + d\,V},$$

or when x_a is a known quantity we may say

$$x = \frac{B}{760}\,x_a.$$

For each fuel-vapor there is required a certain proportion of air for its complete combustion in a gas-engine cylinder. Practice has shown that gasoline requires for effective combustion, approximately, 15 per cent more air than the quantity theoretically

necessary, and, though it has not as yet been fully settled, there is no doubt but that alcohol-fuels should be supplied with an excess of air of about 50 per cent above that required by analysis. Under these conditions, the best mixture for the following fuels would be:

Gasoline, 1 pound to air 17.3 pounds, against 15 pounds by analysis. Denatured alcohol, 1 pound to air 11.7 pounds, against 7.8 pounds by analysis. 50 per cent carbureted alcohol, 1 pound to air 15.9 pounds, against 10.6 pounds by analysis.

In the following table are computed the vapor-pressure of some fuels in mixture with such proportions of air as will be theoretically, or practically, required for complete combustion:

TABLE XIV *A*

Fuel.	Proportion Air.	*V.*	*d.*	x m/m	Critical Temp'rature Degrees F.
Gasoline	Am't. req'd. by a'ly's	186	0.24	16.6	....
Gasoline	1.15× " " " "	214.0	.....	14.5	....
Ethyl alcohol, pure	" " " "	112.7	0.129	49.2	+72
Methyl alcohol, pure	" " " "	80.6	0.089	93.0	68
Denatured alcohol 90%	" " " "	96.7	0.101	70.6	78
Denatured alcohol, 90%	1.50× " " " "	145.1	.....	48.6	67
50% Carbureted alcohol	" " " "	131.4	0.162	34.1	24
50% Carbureted alcohol	1.50× " " " "	197.2	.....	23.0	12

In the last column is given the minimum temperature at which the mixture can exist, at atmospheric pressure, without the fuel becoming condensed. These temperatures are taken from Tables VIII and XIV.

The preceding table shows that 48.6 $^m/_m$ mercury is the vapor-pressure of the fuel in a suitable mixture of denatured alcohol and air, and 67° F. is the minimum temperature at which such a mixture can exist and still retain the full charge of the fuel in vapor, under 760 $^m/_m$ barometric pressure. It also shows that the vapor-pressure of a 50 per cent carbureted alcohol-fuel in a suitable mixture, and at 760 $^m/_m$ barometric pressure, is 23.0 $^m/_m$ mercury, and the minimum temperature at which it

can exist is 12° F. The air supply in both cases being figured 50 per cent in excess of that required by analysis.

When a fuel-mixture is under a partial vacuum, as it will be during the suction stroke in an engine-cylinder, then its vapor-pressure and the corresponding critical temperature become lower. If it be assumed that the minimum suction-pressure is 13.2 pounds, or 684 $^m/_m$ mercury, then the vapor-pressure of denatured and of 50 per cent carbureted alcohol, in a mixture including 50 per cent excess air, becomes, respectively,

$$\frac{13.2}{14.7} \times 48.6 = 43.7 \ ^m/_m \text{ and } \frac{13.2}{14.7} \times 23 = 20.7 \ ^m/_m,$$

and the corresponding critical temperatures will be 63° and 9° F.

The vapor-pressure of a mixture not including any excess air would, in other respects under the same conditions, be for denatured alcohol 63.5 $^m/_m$ and for 50 per cent carbureted alcohol 30.7 $^m/_m$, and the corresponding critical temperatures, respectively, 75° and 18° F.

The Minimum Initial Temperature of the Air and Alcohol Charge.—If it should be required that the fuel were completely vaporized in the carbureter before being admitted to the cylinder, then the latent heat of vaporization may be assumed to be all supplied from the sensible heat of the fuel and air, which would practically be the case if the vaporization were to take place in a common unheated carbureter.

The latent heat of vaporization from 62° F. of denatured alcohol is approximately 525 B.T.U. per pound, and that of 50 per cent carbureted alcohol is 350 B.T.U. That the latent heat of the former fuel is considerably greater than that of the latter is due to the greater percentage of water present, whose latent heat is high.

The specific heat of the gases from the two fuels may be assumed, according to the estimate, page 155, to be:

That of denatured alcohol-gas = 0.455
That of carbureted alcohol-gas = 0.374
and that of air is approximately = 0.24.

Assuming the allowance of air per pound of fuel to be:

	Air theoretically required.	Air including 50% excess.
For denatured alcohol .	7.8	11.7
For carbureted alcohol .	10.6	15.9
The specific heat of the gas-and-air mixture becomes then,		
For denatured alcohol .	$\frac{0.24\times 7.8+0.455}{8.8}=\frac{2.327}{8.8}$	$\frac{0.24\times 11.7+0.455}{12.7}=\frac{3.263}{12.7}$
For carbureted alcohol .	$\frac{0.24\times 10.6+0.374}{11.6}=\frac{2.918}{11.6}$	$\frac{0.24\times 15.9+0.374}{16.9}=\frac{4.19}{16.9}$
And the fall in the temperature of the mixture, due to the abstraction of heat for the vaporization of the gas,		
For denatured alcohol .	$\frac{525}{2.327}=226°F.$	$\frac{525}{3.263}=161°F.$
For carbureted alcohol .	$\frac{350}{2.918}=120°F.$	$\frac{350}{4.19}=83°F.$
Hence, to effect a complete vaporization of the fuel, the air and fuel should be initially of a temperature of, at least,		
For denatured alcohol .	226+75=301°F.	161+63=224°F.
For carbureted alcohol .	120+18=138°F.	83+9 = 92°F.

By early engine-trials it was found that alcohol-fuels could not be utilized without pre-heating the charge considerably, and in many cases it was found impossible to start an engine on a cold alcohol-charge; therefore, in such cases, gasoline was resorted to as a means for starting and heating up the alcohol-engine. It appears also evident from the preceding estimate of the critical temperatures, that, in order to vaporize an alcohol-fuel (particularly denatured alcohol) completely in the carbureter, it is required that the air be highly pre-heated.

It will be practical, however, to vaporize the fuel, only partially, in the carbureter and to allow complete vaporization to take place when, during the suction-stroke, the new charge is

mixed with the hot neutrals from the preceding charge. By late experiments it has been found that, while some pre-heating is favorable for complete combustion and high efficiency, it will not be necessary, or desirable, to heat the air to the temperature that a complete vaporization of the fuel in the carbureter would call for.

In some instances the pre-heating of the air to only some 150° F., though the compression used was quite moderate, gave rise to self-ignitions, which, of course, on account of the uncertainty in regard to the exact point of the stroke where ignition may occur, must be considered objectionable. With a properly constructed and cooled combustion-chamber any difficulty on this score should, however, readily be avoided.

To make it possible to employ a very high compression and obtain high efficiency, some builders inject water to the cylinder with the fuel. This is a feature of the Banki, the Deutz and the Marienfelde oil and alcohol engines, tests of which are recorded in Table XXXI, page 410.

The alcohols vaporizing much more slowly than gasoline, it is, with respect to them, of greater necessity to effect a thorough mixture between the air and fuel in the carbureter, and to provide ample port-areas, so as to insure a slow fluid velocity through the carbureter and intake-chambers, thereby avoiding any considerable pressure-reduction in the cylinder at the suction-stroke.

Denatured Alcohol as Fuel.—Practice showing that any high pre-heating of the charge is undesirable, it may be assumed, for the sake of an estimate of the required cylinder capacity of an alcohol-engine, that the vaporization of the fuel is completed first at the time of the beginning of the compression-stroke, and that the temperature of the charge, at that time, is consistently low.

Carrying out, then, the estimate with reference to a denatured alcohol-fuel, the temperature of the charge at the beginning of the compression should not be less than 63° F., at a pressure of $684^m/_m$ mercury, and the volume of the expanded charge per pound of fuel becomes

$$\frac{V_a}{V_o}(x\,a + 1) = \frac{760}{684}\,\frac{523}{522}\quad(x\,a + 1) = 1.1\;(x\,a + 1).$$

Assuming that the higher calorific power of a denatured alcohol

fuel is 11,700 B.T.U., its lower calorific value, after deducting the latent heat in about 1.15 pound of water-vapor formed at its combustion, will be, approximately, 10,450 B.T.U. The air necessary for its complete combustion is, theoretically, 7.8 pounds per pound of fuel, or 102.5 cubic feet at 62° F., and atmospheric pressure.

Each pound of fuel, in vapor, at this temperature and pressure occupying 9.4 cubic feet, we get:

The volume of air necessary according to analyses, per cubic foot of fuel-vapor . . . $a = 10.9$ cubic feet.

Add 50 per cent excess air . . . . . 5.45 cubic feet.

The volume of air to be supplied per cubic foot of fuel-vapor . . . . . $xa = 16.35$ cubic feet.

The total volume of the mixture, per cubic foot of fuel-vapor . . . $xa + 1 = 17.35$ cubic feet.

The volume of the expanded normal charge after completed suction-stroke; using the value $\frac{V_a}{V_o} = 1.1$

$$\frac{V_a}{V_o}(xa + 1) = 1.1 \times 17.35 = 19.08 \text{ cubic feet.}$$

The heating-value per cubic foot of vapor $H = 1111$ B.T.U.

The heating-value per cubic foot of the expanded normal charge

$$\frac{H}{\frac{V_a}{V_o}(xa + 1)} = \frac{1111}{19.08} = 58 \text{ B.T.U.}$$

With a compression ratio of 7 to 1, that should be used in an alcohol-engine, there should, according to Table IV, be realized an efficiency approximately 0.31. In practice, this efficiency has been well exceeded under good average conditions. Counting, therefore, on an efficiency 0.31, the minimum required suction-displacement per minute, per I.H.P., will be

$$D_1 = \frac{42.42}{Efy \times \frac{H}{\frac{V_a}{V_o}(xa + 1)}} = \frac{42.42}{0.31 \times 58}$$

$$D_1 = 2.36 \text{ cubic feet per minute.}$$

The required suction-displacement per minute per rated I.H.P., allowing 15 per cent overload capacity, is

$$D_2 = 1.15 \frac{42.42}{0.31 \times 58} = 2.71 \text{ cubic feet per minute.}$$

The required suction-displacement per minute per rated B.H.P., assuming the mechanical efficiency to be 0.85, is

$$D_4 = 1.35 \frac{42.42}{0.31 \times 58} = 3.19 \text{ cubic feet per minute.}$$

The capacity of an engine of given dimensions, when operating on denatured alcohol-fuel, will be, in B.H.P.,

$$\text{B.H.P.} = \frac{l\,a\,N}{3.19 \times 3{,}545} = \frac{l\,a\,N}{11{,}300}$$

l being the length of the stroke in inches, a the area of the piston in square inches, and N the number of revolutions per minute.

The mean effective pressure on which the required suction-displacement is based will, according to equation 50*b*, be

$$\text{M.E.P.} = 98 \text{ pounds.}$$

CHAPTER VIII

FEATURES OF THE PRACTICAL GAS-ENGINE CYCLE

Ignition.—In order to explode a fuel-mixture it is required only that a small portion of the fuel be heated to the ignition-temperature. The heat of combustion of only a small portion of the explosive mixture will cause the inflammation of the whole charge, either through ordinary flame-propagation or possibly by means of an explosive wave which is assumed to cause practically instantaneous combustion of the whole charge.

The ignition-temperatures, which have been established by various experimenters, for hydrogen, carbon monoxide, marsh gas and ethylene are approximately:

For H	mixed with O	1100° Fahr.
For CO	mixed with O	1300° Fahr.
For CH_4	mixed with O	1200° Fahr.
For C_2H_4	mixed with O	1100° Fahr.

The diluting of a mixture does not generally change its ignition-point, excepting in the case of CO gas mixed with CO_2, in which case the ignition-point becomes materially higher. The ignition of the various gases in air occurs, thus, approximately at the same temperature as when mixed with oxygen.

The Timing of the Ignition.—The ignition of the charge should be timed to suit the fuel, the compression, and the speed of the engine. The surest means for finding the best point for ignition is the indicator, but it can generally be determined approximately by listening to the sound in the engine while the timing-device is varied.

In Fig. 37, I, II and III, are represented cards from an engine running on natural gas, taken at full load with the ignition timed differently in each case, and they show:

Card I a correct ignition.

Card II a too early ignition.

Card III too late ignition.

Card IV is taken at a lighter load and with an ignition entirely too late to suit the low compression of the charge.

Flame Propagation.—The rate of the flame propagation in a perfectly vaporized explosive mixture of normal composition is very high, and ordinarily no essential advantage would be derived

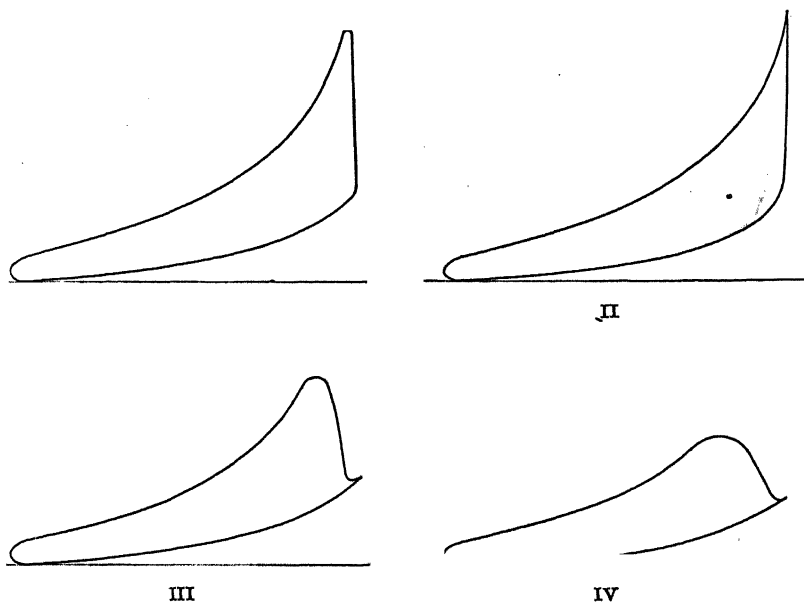



FIG. 37.

by an increased rate. In a diluted mixture, however, the rate is often not what would be desired.

Experiments by Mr. Grover,* at Yorkshire College, show that the replacing of the excess air in a diluted coal-gas mixture, in part, with burned gases (neutrals), may increase the rapidity of its combustion. Experimenters are not united, however, as to what influence the neutrals have on the rapidity of combustion of an explosive mixture, in general, though with respect to some fuels the detriment of replacing excess air for neutrals seems to be proven.

* "Modern Gas and Oil Engines," by Frederick Grover.

Mr. Grover's experiments establish the following additional conclusions with respect to a coal-gas mixture:

That the highest pressures are obtained when the volume of air is only slightly in excess of the amount required for complete combustion.

That when the volume of the products of combustion does not exceed 58 per cent of the total mixture then the mixture is still explosive, provided the volume of air is not less than 5½ times the volume of gas.

That higher pressures are recorded when the residual gases take the place of excess air.

The above conclusions, with regard to the experiments, amount practically to the same as saying that a perfect coal-gas-and-air mixture may be diluted with 140 per cent of its volume of burned gas and still be explosive; and the velocity of the flame-propagation of such a mixture is more rapid than if air were used for diluent.

The temperature of the charge when ignition takes place has also influence on the rate of the flame-propagation, so that the higher the temperature the more rapid the combustion becomes.

Experience also shows that mixtures in which the fuel is imperfectly vaporized often give unsatisfactory results due to slow combustion, and that a charge too weak to explode with a normal spark can be made to explode by using a heavy spark.

Explosion Experiments.—Clerk's experiments on explosive mixtures are of interest as showing the rapidity with which combustion occurs.

These experiments were made with Glasgow and Oldham city-gas with varying proportions of air. The ignition of the charge was effected without previous compression, at 60° F., and the results are shown in Table XV. The highest pressure attained was, as will be seen, 91 pounds per square inch.

Some experiments on mixtures of hydrogen and air are given in Table XV *a*.

For Koerting's experiments with coal-gas under a moderate pressure see Table XV *b*.

TABLE XV

Proportion by Volume.		Maximum Observed Pressure. Pounds per sq. in.	Time to Reach Maximum Pressure. Seconds.
Gas.	Air.		
1	14	40	0.45
1	13	51.5	0.31
1	12	60	0.24
1	11	61	0.17
1	10	.....	.....
1	9	78	0.08
1	8	....	.....
1	7	87	0.06
1	6	90	0.04
1	5	91	0.055
1	4	80	0.16

TABLE XV *a*

Proportion by Volume.		Maximum Observed Pressure. Pounds per square inch.	Time to Reach Maximum Pressure. Seconds.
Hydrogen.	Air.		
1	6	41	0.15
1	4	68	0.026
2	5	80	0.01

TABLE XV *b*

Proportion by Volume.		Pressure Before Ignition. Pounds per sq. in. Absolute.	Time to Reach Maximum Pressure. Seconds.	Velocity of the Propagation of Pressure. Ft. per second.
Gas.	Air.			
1	7.5	15.0	0.032	23.0
		37.0	0.036	20.4
1	5.42	15.0	0.01	44.0
		37.0	0.0125	59.0

The propagation of the flame in the experiments by Koerting is essentially higher than in those of Clerk, which undoubtedly is due to differences in the general conditions under which the experiments were made. Comparing each set of results in Table XV *b* it will be seen that the flame-propagation is somewhat more rapid at the low pressures than at the higher ones. This may be according to what would be expected, as there is more weight of gas to be consumed in the latter case than in the former. The consumption of a unit weight of gas is, however, much more rapid at the higher pressure.

The following tables XVI and XVII are the results of experiments on explosive mixtures made at the Massachusetts Institute of Technology, and in Fig. 37 *a* are constructed, on a time basis, the pressure-lines obtained from every second of the mixtures of Table XVI; beginning with the 1 to 3 mixture.

TABLE XVI.

Results of Tests on Explosive Mixtures of Illuminating-Gas and Air.

Mixi (by Vol	Maxi ure. Pou in.	Expl cond:	Sq	First ⅛ Second. Mean Pressure. Pounds per sq. in.	Mean Pressure ÷ Proportion of Gas.	Final P	⅛ Second after Maximum Pressure. Area. ire inc	Mean Pound	Mean Pressure ÷ Proportion of Gas		Pressure ÷ rtion of Gas.
Gas-air											
1–3	45	.49	0.32	11	44	26	1.80	43	172	40	160
1–4	86	.08	1.77	59	295	61	1.88	62	310	46	230
1–5	96	.05	1.86	62	372	52	1.93	64	384	44	264
1–6	88	.05	1.80	60	420	54	1.93	64	448	46	322
1–7	86	.06	1.97	66	528	58	1.93	64	512	46	384
1–8	87	.06	1.71	57	513	53	1.83	61	549	46	414
1–9	77	.08	1.60	53	530	57	1.86	62	620	46	460
1–10	71	.11	1.36	45	495	56	1.69	56	616	45	495
1–11	68	.14	1.21	40	480	60	1.66	55	660	43	516
1–12	39	.33	0.35	12	156	29	0.98	33	429	30	390
1–13	32	.42	0.18	6	84	16	0.79	26	364	24	336
1–14	9	.42	0.05	2	30		0.24				120

It will be noticed by the diagrams, Fig. 37 *a*, that, generally, the quicker the combustion occurs, and the higher the initial pressure, the lower the pressure-line becomes after one second's time, excepting in the case of the very rich mixture, curve 3. The end of the first one-fifth second from the time of the ignition is marked in the figure by a heavy line at the point $\frac{12}{60}$ of the total length of the diagrams, and at that line the final pressures

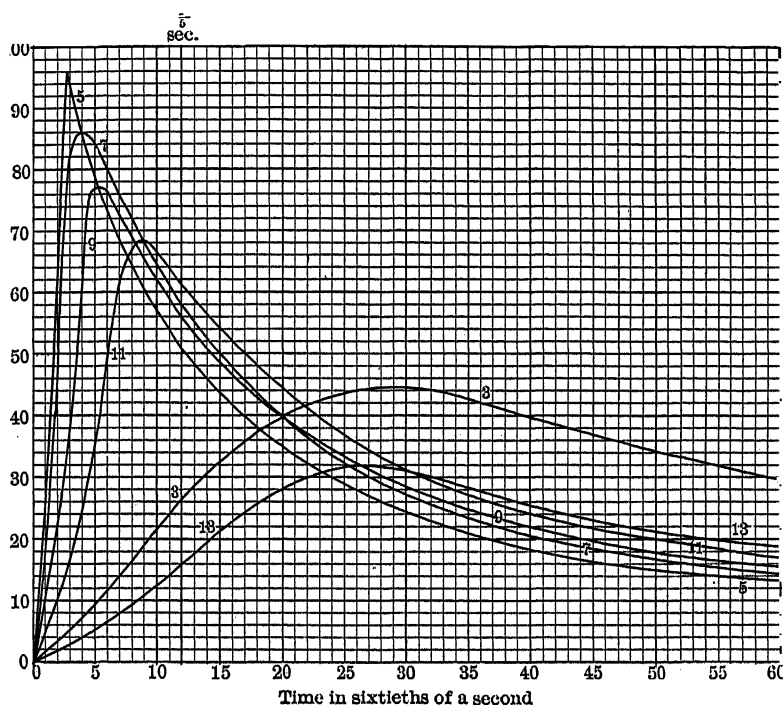



FIG. 37 *a*.

after the first one-fifth second are measured. The areas obtained after the first one-fifth second, and the mean pressures, columns 4 and 5 of Table XVI, refer also to the areas of the diagrams up to that line. The figures of the sixth and tenth columns of the tables represent, of course, the efficiencies at the combustion of each of the mixtures.

TABLE XVII.

Results of Tests on Explosive Mixtures of Gasoline (76°, Sp. gr. 0.680) and Air.

Proportion of Gas in Mixture.	Time of Explosion. Seconds.	First 0.2 Second.				Maximum Pressure.	0.2 Second after Max. Pressure.			
		Area. Square inches.	Mean Pressure. Pounds per sq. in.	Mean Pressure ÷ Proportion of Gas.	Final Pressure.		Area. Square inches.	Mean Pressure. Pounds per sq. in.	Mean Pressure ÷ Proportion of Gas.	Final Pressure.
0.0132	.167	.76	25.3	1925	52	52	1.28	42.7	3240	33
0.0141	.117	1.15	38.4	2720	49	62	1.42	47.3	3360	35
0.0151	.109	1.26	42.0	2770	48	64	1.45	48.6	2950	35
0.0164	.182	.81	27.0	1650	50	51	1.25	41.7	2540	32
0.0179	.109	1.27	42.3	2368	50	67	1.53	51.0	2855	36
0.0196	.091	1.44	48.0	2441	48	73	1.53	51.0	2600	36
0.0217	.082	1.43	47.7	2180	48	76	1.56	52.0	2391	37
0.0244	.060	1.62	54.0	2213	45	85	1.63	54.3	2225	36
0.0263	.058	1.61	53.7	2040	45	85	1.62	54.0	2052	36
0.0278	.058	1.62	54.0	1943	46	84	1.64	57.4	1970	38
0.0303	.066	1.49	49.7	1640	45	78	1.60	53.4	1760	37
0.0323	.067	1.55	51.7	1602	48	83	1.70	56.7	1760	38
0.0345	.100	1.34	44.7	1297	52	75	1.59	53.0	1536	38
0.0385	.117	1.10	36.7	955	52	62	1.42	47.3	1230	35
0.0417	.133	.98	32.7	761	52	55	1.40	46.7	1121	38
0.0476	.210	.39	13.0	273	35	35	1.02	34.0	714	32

Mixtures Highly Diluted with Combustion-Products.—It may appear from Mr. Grover's experiments with mixtures diluted with air and burned gases as if the neutrals in a weak mixture would tend to facilitate rather than retard the inflammation of the charge. However, the conditions under which these experiments were carried out were materially different from those existing during the combustion in the gas-engine cylinder, and experience, with some fuels at least, shows results differing in this respect. What effect, then, will the combustion-products actually have on the combustion in the gas-engine cylinder?

Assume a case of a throttling engine of fairly high compression,

which, due to faulty valve-setting, does not relieve itself of the burned gases to the best advantage. Such an engine we find, on a heavy load, apt to give evidence of pre-ignitions, which are due probably to the excess heat the burned gases transmit to the charge. Under a light load, however, the operation of the engine is liable to be troubled by back-firing into the mixing-chamber or inlet valve-casing, at times of the opening of the inlet value for admission of the new charge. This feature would be explained on the ground that an excessive amount of combustion-products in a weak mixture makes it slow-burning, to the extent of holding the fire all through the cycle until the inlet valve begins to open for a following suction-stroke.

After readjusting the valve-setting, by opening the exhaust valve early and keeping it open as long as practical, an appreciable change will have been accomplished. The cylinder will become scavenged from combustion-products as far as possible; hence the weak charge, on the same light load as before, will be diluted by more air instead of by an excess of combustion-products, and the back-firing into the mixing-chamber will have been cured. This tends to show, that of the two mixtures of similar heating-value the latter, containing less combustion-products, acts, in the working cylinder, as if it were the more inflammable.

Suppression of Heat at Combustion.—The specific heats of gases have been found to be approximately constant for temperatures as high as 500 degrees Fahr., and it has generally been assumed to be so for all temperatures. The heating-value admitted to the gas-engine cylinder with each pound of charge being known, the increase in temperature through combustion should, according to this assumption, theoretically be

$$t_f - t_i = \frac{Q}{c_v}.$$

However, judged by the corresponding increase in the pressure, this increase in temperature is never realized in practice, and a phenomenal suppression of from 30 to 40 per cent of the total heating-value of the fuel is often observed.

There has existed, and still exists perhaps, to some extent,

differences of opinion as to the cause of this discrepancy between the maximum temperature that should theoretically be obtained at the combustion in a gas-engine cylinder and the temperature actually obtained. One of the theories regarding its cause assumes that the metal of the combustion-chamber absorbs the heat apparently lost, later restoring part of it to the charge during its expansion; the effect being similar to that due to the so-called "wall-action" in a steam-engine cylinder.

The cause has, by another theory, been sought in the fact that some of the combustion-products become dissociated at very high temperatures, and that, therefore, the temperature obtained must always be below that at which dissociation takes place. According to a third theory, the suppression of heat is due to slow combustion, which would have for effect the lowering of the temperature through expansion before the full heating-value becomes evolved by combustion.

That the wall-action of the cylinder has some influence on the temperature obtained at the combustion there is no doubt, but, as a following approximate determination of the same will tend to show, it cannot be very considerable. It has, however, a decided tendency to absorb a certain amount of heat at the early part of the explosion-stroke, and to restore it, partly, later during the expansion. Clark's dissociation theory has lately been proven incorrect, on the ground that the temperature in the cylinder never attains that intensity at which either the carbonic acid gas or steam are dissociated. Finally, the after-burning theory, although it has full force with respect to certain slow-burning mixtures, does not seem to be true with respect to mixtures giving an average good combustion. Entropy-temperature diagrams constructed from indicator cards showing proper combustion-lines do not exhibit from the time of maximum pressure any additional heat-energy above what would readily be due to the action of the cylinder walls.

At present, when it appears certain that the specific heat of all gases formed at the combustion in the cylinder increases with the temperature, the so-called "heat suppression" would seem to have found a solution. And, with the assumption that the

values of the specific heat of gases at high temperatures which are at hand up to date are approximately correct, it can readily be ascertained that the temperatures and pressures obtained in the gas-engine are not materially lower than what can be expected theoretically.

According to determinations made by Mallard and Le Chatelier, the specific heats for the gases quoted below increase with the temperature in the following rate; the temperature being expressed in degrees Fahr.

TABLE XVIII

The specific heat at constant pressure.

$$
\begin{aligned}
CO_2 &= 0.185 + 0.000093\,t \\
H_2O &= 0.415 + 0.000202\,t \\
N &= 0.240 + 0.000024\,t \\
O &= 0.211 + 0.000021\,t \\
\text{Air} &= 0.233 + 0.000023\,t
\end{aligned}
$$

The specific heat at constant volume.

$$
\begin{aligned}
CO_2 &= 0.140 + 0.000093\,t \\
H_2O &= 0.306 + 0.000202\,t \\
N &= 0.173 + 0.000024\,t \\
O &= 0.150 + 0.000021\,t \\
\text{Air} &= 0.165 + 0.000023\,t
\end{aligned}
$$

Assume that in an engine there is used as fuel gasoline, the lower heating-value of which is 18,500 B.T.U., and that the compression ratio is $r = 4$.

Thus, $r^{n-1} = r^{0.35} = 1.62$.

The normal expanded charge after completed suction-stroke, including 15 per cent excess air, contains under these conditions, at about 80 degrees F., the heating-value (see page 127)

$$\frac{H}{\frac{V_a}{V_o}(x\,a + 1)} = 70 \text{ B.T.U. per cubic foot.}$$

Hence, the heating-value per pound of normal charge will be, according to equation 44,

$$Q = \frac{H}{\frac{V_a}{V_o}(x a + 1)} V_a \frac{r - 1}{r} = 70 \times 13.5 \times 3/4 = 708 \text{ B.T.U.}$$

The maximum temperature obtained at the combustion has been found by investigators to be, ordinarily, between 3,000° and 3,800° Fahrenheit—on an average thus 3,400° Fahrenheit.

The absolute temperature after compression is

$$T_b = r^{n-1} T_a = 1.62 (460 + 80) = 875° \text{ F.}$$

or 415° above Fahrenheit zero.

The mean temperature during the combustion may, therefore, be called, approximately 1,900° F.

If this temperature, 1,900° F., be inserted in the preceding equations for the specific heat at constant volume, for CO_2, H_2O, N and O, we obtain the following:

The mean specific heat at constant volume for a range of temperatures between 415° and 3,400° F., for

	S_v
CO_2	= 0.317
H_2O	= 0.690
N	= 0.218
O	= 0.190

At the combustion of one pound of gasoline in 15 per cent excess air, there are formed the following products:

CO_2	3.0 pounds
H_2O	1.4 pounds
N	13.0 pounds
O	0.5 pounds
Total	17.9

The average specific heat at constant volume, S_v, of all the products of combustion, therefore, for a range of temperatures between 415° and 3,400 F., is

Combustion Products, lbs.		S_v		
3.00	×	0.317	=	0.951
1.40	×	0.690	=	0.966
13.00	×	0.218	=	2.834
0.50	×	0.190	=	0.095
				4.846

$$S_v = \frac{4.846}{17.9} = 0.27$$

Inserting the preceding values for Q, S_v and $r^{n-1} T_a$ in equation 43, which may be written:

$$\frac{P_c}{P_b} = 1 + \frac{fQ}{S_v r^{n-1} T_a}, \text{ we obtain}$$

$$\frac{P_c}{P_b} = 1 + \frac{f}{0.27} \; \frac{708}{875} = 1 + 3.00 f.$$

Assuming that during the combustion there will be no loss of heat to the cylinder walls, thus $f = 1$, then

$$\frac{P_c}{P_b} = 4.00$$

In a non-scavenging engine this value, $\frac{P_c}{P_b} = 4$, is obtained very rarely, and for ordinary running a very good result would be $\frac{P_c}{P_b} = 3.75$, which corresponds to an initial pressure 304 pounds above the atmosphere.

Assuming $\frac{P_c}{P_b} = 3.75$ to be a normal value for good cards, then f becomes 0.917, showing a loss to the combustion-chamber walls of 8.3 per cent of the total heating-value.

The maximum temperature at the combustion will be obtained from equation $f\,Q = S_v (T_c - T_b)$, which for $f = 0.917$ becomes

$$0.917 \times 708 = 0.27 \, (T_c - 875).$$

Thus, $$T_c = 3280° \text{ F.}$$

This value is near enough to the assumed value $T_c = 3400°$ F., on which the computation for the mean S_v was based, to make it unnecessary to carry through any correction.

Heat-loss at the Combustion.—It may be possible to get an approximate idea about the amount of heat that reasonably can be expected to dissipate into the metal of the combustion-chamber, during the short time the combustion is in progress, through comparison with the total heat that becomes dissipated into the water-jacket during the time of the whole cycle.

Assume a 24 × 36 single-acting engine of a piston speed of 600 feet per minute. Thus, 100 revolutions per minute, or 1⅔ revolution per second.

The indicated power of this engine is

$$\frac{A\,S}{4\,D_1} = \frac{314 \times 600}{4 \times 2.55} = 180 \text{ I.H.P.}$$

Hence the heating-value transferred into indicated horse-power is:

$$\begin{aligned} &\text{per minute } 180 \times 42.42 &= 7636 \text{ B.T.U.,} \\ &\text{per second} &= 127 \text{ B.T.U.,} \\ \text{or } &\text{per cycle} &= 152.72 \text{ B.T.U.} \end{aligned}$$

From tests it is known that in an engine of this class there is dissipated into the water-jacket, generally, heat amounting to somewhat more than that transformed into indicated power. Assume that 10 per cent more heat is dissipated than that obtained as indicated power. Thus, a total of 168 B.T.U. per cycle.

Add to this heat-energy again 10 per cent of the heating-value accounted for by the indicator card, or 15 B.T.U., to allow for the heat which is returned to the charge during the later part of the expansion-stroke. This percentage is ample as, actually, entropy-temperature diagrams generally show not much over one-half of this amount.* Add also the heat which the incoming charge absorbs, from the cylinder, which, according to page 113, is about 24,300 B.T.U. per hour, or, 8 B.T.U. per cycle.

168 + 15 + 8 (= 191) B.T.U. is, accordingly, abstracted from the working gases, during the latter part of the compression-stroke, during the explosion and during the beginning of the expansion-stroke. During the latter part of the expansion-stroke

* Compare "Entropy Analysis of the Otto Cycle." S. A. Reeve, Transact. Am. Soc. Mech. Engs., Vol. XXIV.

and during the exhaust-stroke the gases are of such an essentially lower temperature than at the early part of the expansion that no heat can then pass from them into the metal—the opposite might rather be the case, at least during part of the period.

Assume, therefore, that all the heat is abstracted during one-half of one revolution, or during $\frac{1}{2} \times \frac{3}{5} = \frac{3}{10}$ second. Thus, 191 B.T.U. is abstracted in $\frac{3}{10}$ of a second, or at the mean rate of 637 heat-units per second.

The mean temperature of the gases during the combustion is, of course, practically the same as their mean temperature during the expansion, but, the metal of the combustion-chamber being at the former period comparatively cold, it is to be assumed that the rate at which heat is abstracted then is much greater than during the time of expansion. A fair assumption would, probably, be that the rate at which heat is abstracted during the combustion is twice the mean rate at which it is abstracted during the entire period of heat-loss.

At that rate there would be abstracted during the combustion 1,274 heat-units per second, and as the average time for a proper explosion is, say, 0.04 of a second, the total amount of heat dissipated during the explosion becomes 50 heat-units. Assuming the thermal efficiency to be 25 per cent, then the heat dissipated into the metal of the combustion-chamber becomes $\frac{50 \times 0.25}{152.72} =$ 8 per cent of the total heating-value.

The above results are, of course, founded on several assumptions that cannot very well be proven to be exactly correct, but if the assumptions are reasonable the results will tend to show that the heat-suppression observed, in connection with any good combustion in the cylinder, can very well be explained, since the specific heat increases materially at high temperatures, and since the metal of the combustion-chamber actually absorbs, during the combustion, from 5 to 10 per cent of the total heating-value liberated.

The Effect on the Expansion-Line of the Diluting of the Charge.—In Fig. 38 are drawn three curves, the first an isothermal curve, the second and third adiabatics for $n = 1.25$ and for

$n = 1.3$. It is evident that the smaller the ratio $\frac{c_p}{c_v} = n$ is the nearer to the isothermal line the expansion line will fall, and the greater the M.E.P. and area of the indicator card becomes.

In a card from a diluted mixture, the mean effective pressure is often higher, relatively to the initial pressure, than in a card from a normal mixture, and this may be thought to be due to the difference in the specific heat of the combustion-products from the diluted mixture and from the normal mixture. If it be assumed, however, that the specific heats of the elementary gases increase with the temperature in the rate Table XVIII, page 173, shows then this cannot be the case.

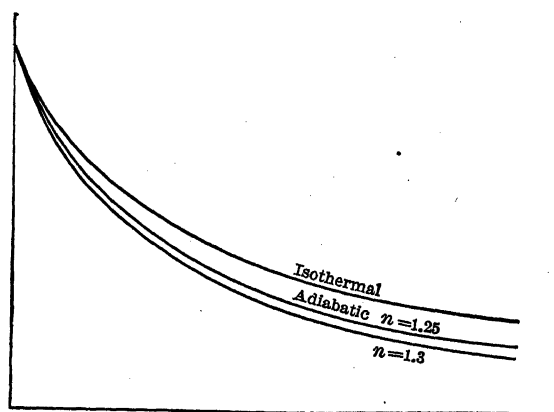



FIG. 38.

Computing, similarly as on page 175, the specific heats of the products of combustion from gasoline, including 15 per cent excess air, at the mean temperature 1,900° F., we obtain the following:

	Combustion Products, lbs.	Combustion Products, lbs. c_v
CO_2	$3.0 \times 0.362 = 1.086$	$3.0 \times 0.317 = 0.951$
H_2O	$1.4 \times 0.799 = 1.118$	$1.4 \times 0.690 = 0.966$
N	$13.0 \times 0.286 = 3.718$	$13.0 \times 0.218 = 2.834$
O	$0.5 \times 0.251 = 0.125$	$0.5 \times 0.190 = 0.095$
	17.9 6.047	17.9 4.846

$$c_p = \frac{6.047}{17.9} = 0.338 \qquad c_v = \frac{4.846}{17.9} = 0.27$$

$$\frac{c_p}{c_v} = 1.25 = n.$$

On the other hand, assume that the mixture be diluted with the same quantity of air that is actually required for its combustion; it accordingly being charged with about 15 pounds excess air. The mean temperature at the combustion will then be considerably less than before, approximately only 1,500° F.

The computation for the specific heat of the products of combustion of gasoline-fuel diluted with 100 per cent excess air, at the mean temperature 1,500° F., will be:

	Combustion Products, lbs.	c_p		Combustion Products, lbs.	c_v	
CO_2	3.0	× 0.324	= 0.972	3.0	× 0.279	= 0.837
H_2O	1.4	× 0.718	= 1.005	1.4	× 0.609	= 0.853
N	11.6	× 0.276	= 3.202	11.6	× 0.209	= 2.424
Air	15.0	× 0.267	= 4.005	15.0	× 0.199	= 2.985
	31.0		9.184	31.0		7.099

$$c_p = \frac{9.184}{31} = 0.296 \qquad c_v = \frac{7.099}{31} = 0.229$$

$$\frac{c_p}{c_v} = 1.3 = n.$$

As the lowest of the curves, Fig. 38, represents the expansion-line for a diluted charge, it is evident that the change in the specific heat of the combustion-products, due to the diluting of the charge, does not have for effect the raising of the expansion-line; rather the contrary. That the expansion-line from a diluted charge often is higher relatively to the initial pressure than the expansion line from a normal charge depends, it must be concluded, on some other cause apart from its dilution.

The Relation between Initial- and Mean Effective Pressure.— The value of the coefficient y is, according to equation 53, page 40,

$$y = \frac{m.e.p.\ (n-1)\ (r-1)}{E\ (p_c - p_b)}.$$

In order to give this formula a practical application, we solve the value of y for each of the following six cards from an engine, under test, working on kerosene and gasoline fuels. The engine, an Otto hit-or-miss, 6¾ × 15½, 15 brake horse-power engine, was arranged with a special vaporizer for using kerosene.

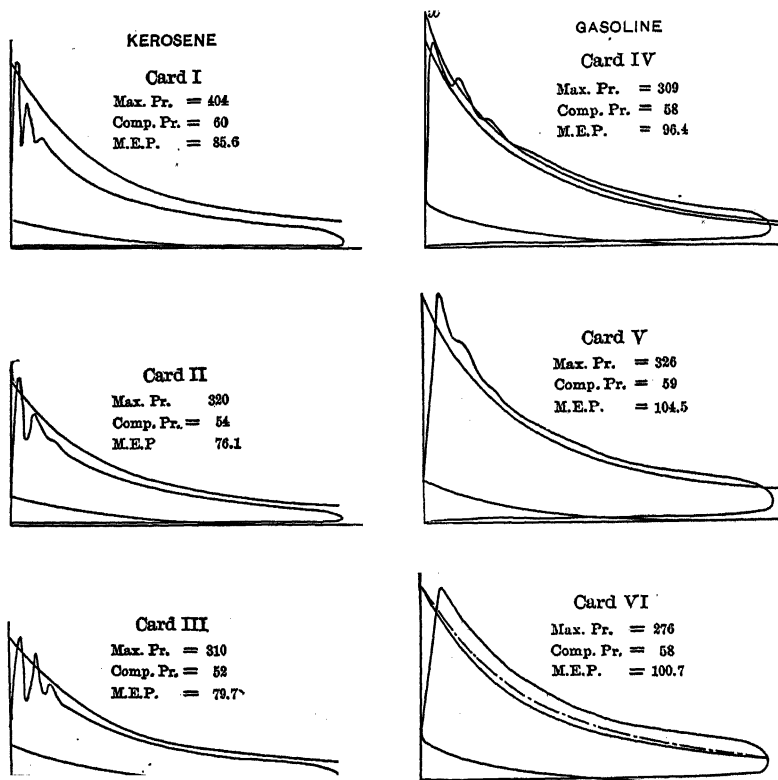



FIG. 39.—CARDS FROM ENGINE-TEST ON GASOLINE AND KEROSENE.

The maximum-, the mean effective-, and the compression-pressures are noted on each card.

The actual compression ratio of the engine is 3.9 to 1, which, compared with the compression pressures obtained, shows that the charge, at the beginning of the compression stroke, was of a pressure somewhat less than that of the atmosphere. We obtain the factor $r - 1 = 2.9$.

The value of the coefficient E, on the basis of a compression ratio 3.9 to 1, is $E = 0.372$ and $n - 1 = 0.35$, approximately.

Hence, the factor $\frac{(n-1)\ (r-1)}{E} = 2.72.$

Inserting the values of $m.e.p.$ and $p_c - p_b$ in the equation $y = 2.72\frac{m.e.p.}{p_c - p_b}$, and solving for y we obtain the following:

	Card.	M. E. P.	pc.	pb.	pc − pb.	y.
Kerosene...	I	85.6	404	60	344	0.7
	II	76.1	320	54	266	0.8
	III	79.7	310	52	258	0.83
Gasoline ...	IV	96.4	309	58	251	1.05
	V	104.5	326	59	267	1.07
	VI	100.7	276	58	218	1.26

If the figures for the cards II and III of the first group be compared we find that, although the initial pressure of card III is the lowest, yet its mean effective pressure is the highest. This we find more in evidence by comparing the figures for cards IV and VI of the second group. VI having, by far, the lowest initial pressure, yet, its mean effective pressure is the highest.

The mean effective pressure of the actual indicator cards we find, thus, essentially independent of the maximum pressure, and the actual area of the card is, as the coefficient y shows, from 30 per cent less to 26 per cent larger than the area of the theoretical card. The variation in the value of the coefficient y we find to be, to some extent, due to the fact that the maximum initial pressure is more or less pushed back away from the head-end of the card.

On each card of Fig. 39 there is drawn an expansion line following the equation $p = \left(\frac{v_1}{v}\right)^{1.35} p_1$, by which the actual and the theoretical expansion lines, as well as the areas of the actual and the theoretical cards, may be compared.

The nature of the fuel, the temperature of the charge, the

proportioning of the mixture, as well as the timing of the ignition, all have influence on the slope of the combustion line, as well as on the appearance of the card at the high point of maximum pressure. Thus we find that the kerosene cards, which were taken with the fuel- and air-mixture arriving to the engine heated to a temperature not less than 600° F., have a much quicker combustion-line, and are sharper at the high point than the gasoline cards, which were taken with the fuel arriving cold and probably less thoroughly vaporized. The expansion lines of the kerosene cards drop, however, quickly below the theoretical expansion line, which evidently is due to the quicker combustion and consequently higher momentary combustion-temperature, whereas the expansion lines of the gasoline cards all show a tendency to keep above this line.

The correctness of the information that the coefficient y will give is not absolute, but it will be close enough for a comparison between cards from different fuels, and particularly so if the cards are all taken from the same engine. It will be apparent, that the coefficient expresses simply the ratio between the mean effective pressure obtained from the actual card and the theoretical mean effective pressure of the air-card. In order, then, that its value shall be reasonably correct it will be required that we shall use in the formula the correct expansion ratio (this ratio is assumed to be the same as the compression ratio) and therefore the true clearance space in the engine must be known.

To illustrate the influence the employment of an incorrect expansion ratio would have on the result given by formula 53, there are in card VI, Fig. 39, drawn two curves, the lower being the correct one for an expansion ratio 3.9 to 1 and the upper for an expansion ratio of 3.5 to 1. The upper curve, enclosing a materially larger air-card area, will give a materially smaller value of the coefficient y. The difference will, however, be of minor importance when the object is simply to compare the different cards on the same basis.

For such cards as shown in Fig. 40, taken on a light load and with the charge throttled, or diluted, considerably, the coefficient y may become very large, but it has in such cases no significance

as the card, then, does not even approximately follow the theory laid down for the cycle. In others, it may be considerably less than the unit, giving evidence of a cold cylinder or leaking valves.

In card IV, through the point x, which, according to the theory, should have been the starting-point for the actual expansion curve, there is drawn a theoretical expansion line ($n = 1.35$). The area below this line is to some extent smaller than the area below the actual expansion curve, and the work represented by the area between the theoretical and actual expansion lines represents the heat added after the beginning of the expansion. This heat is evidently obtained from the heated metal of the

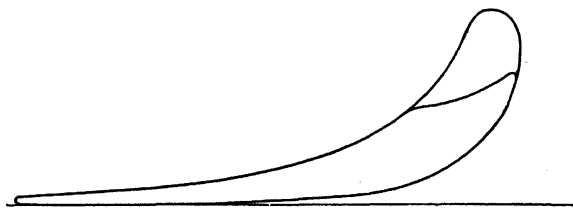



FIG. 40.

cylinder; and the fact that the actual expansion line often rises above the theoretical line toward the end of the stroke, after the cylinder has become well heated, may to some extent help to explain the phenomenon that heat appears to be suppressed at the beginning of the cycle to reappear, later, during the expansion.

At the tests of the kerosene and gasoline fuels, during which the cards, Fig. 39, were taken, the average fuel-consumption, and the efficiency of the heat transformation, were as follows:

	Kerosene.	Gasoline.
Fuel consumption per hour per I.H.P. lbs.	0.675	0.709
Fuel consumption per hour per B.H.P. lbs.	0.873	0.894
Average efficiency for all tests	20.21	17.9
Maximum efficiency at one test	24.89	20.7

The circumstance will be noticed that, although the coefficient y of the gasoline cards, and the cards themselves, appear much more favorable with respect to an efficient utilization of the heat, still, the kerosene gave actually a materially higher efficiency,

which fact tends to show that the individual indicator card, though it tells how effectively the heat was utilized after having been evolved in the combustion-chamber of the cylinder, cannot tell, excepting by comparison with some standard, how effectively the heating-value of the fuel was realized at the combustion.

Explosive Waves.—When indicator cards are taken from gas-

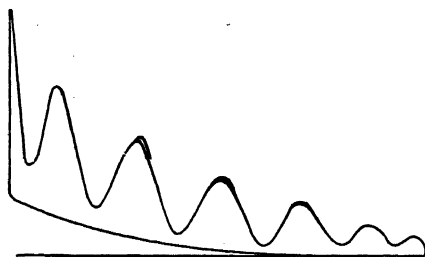



FIG. 41.

engines, there occasionally will appear cards, the expansion line of which shows irregularities that are hard to explain. Sometimes the expansion line, instead of being a smooth curve, becomes a wave-line consisting of a number of undulations, as Fig. 41. Berthelot, the French scientist, explained the phenomenon as

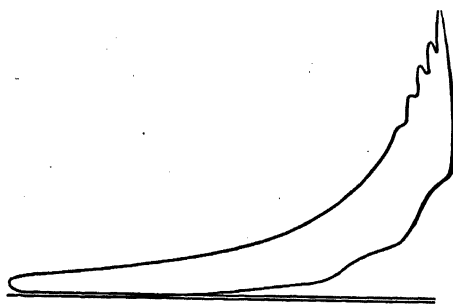



FIG. 42.

being due to, what he called, explosive waves in the cylinder. Investigations have shown that, if the charge becomes agitated during the compression, then the combustion is likely to be followed by violent waves, and experimenters have purposely produced agitation of the charge, by special means, in order to obtain such waves and to verify the cause of them.

In practice the agitation is assumed to originate from undulations in the charge set up through the action of the compression. Compression waves are occasionally observed on the indicator card, as in Fig. 42, and they are sometimes accompanied by more or less violent undulations of the expansion line, as at the

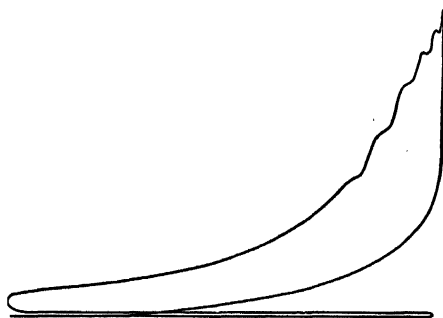



FIG. 43.

top of card, Fig. 42. It cannot be presumed that the waves indicated on the diagram represent, in every case, truly the fluctuations of pressure obtaining in the cylinder. That cannot be, because the inertia of the indicator pencil-arm and mechanism

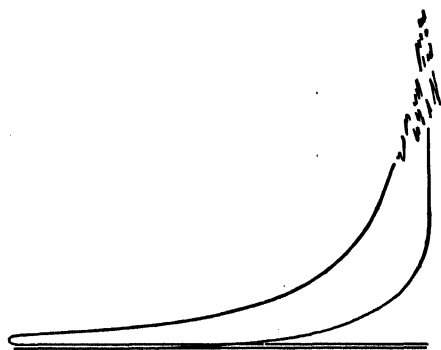



FIG. 44.

must exert a considerable influence on the wave-line indicated, due to the extraordinary rapid motion given to it. Wave-lines, such as shown in Fig. 43, may be entirely due to the inertia of the indicator pencil-arm and mechanism, just as corresponding waves are obtained on the steam-engine card.

There are often vibrations produced on the indicator-card that can readily be obliterated, by effecting an alteration in the connection between the indicator and the inside of the cylinder. Between the combustion-chamber and the tap for the indicator there is often a chamber for the accommodation of the valve that closes the indicator opening to the cylinder, when the instrument is removed. In this chamber there is likely to occur secondary explosions that distort the true expansion line due to the actual pressure in the cylinder, and, to prevent this, a pipe should be fitted, leading from the opening in the cylinder direct to the indicator.

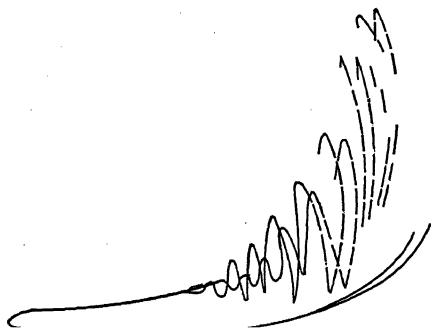



Fig. 45.

Sometimes, when the explosions are powerful, due to a rich mixture and early ignition, the top of the card will consist of a series of vibrations, which are evidently due to the springing of the instrument, and may be aggravated by the springing and vibrating of the metal of the engine forming the combustion-chamber. A card of this nature is shown in Fig. 44.

Occasionally the vibrations of the pencil-arm may be extremely violent, at the beginning of the stroke, and disappear during the expansion, resulting in a card as Fig. 45. This card was taken during two explosions and represents evidently a combination of vibrations due to pre-ignition and explosion waves.

This subject of explosion waves is not, at present, very fully understood, and engineers are at work trying to solve the mystery surrounding the matter. It is, however, apparent that the variety of explosion waves represented by the indicator is not due to a single cause only.

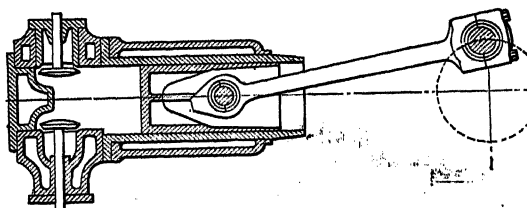
CHAPTER IX

THE FLY-WHEEL

Four-Cycle Engine-Types.—The four-stroke cycle, or four-cycle, engine is built single-acting or double-acting, and is often arranged with two or more cylinders working on a common shaft.

With respect to cylinder arrangement, the following nine types are common for medium-sized and large stationary engines. Types II, III and VI are, however, less frequently used than the others.

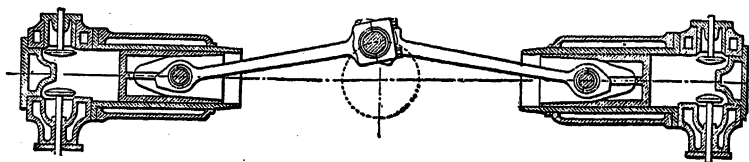
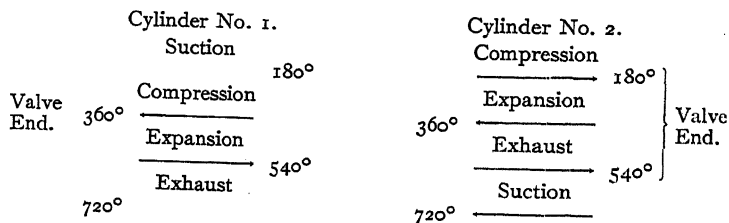
TYPE I

SINGLE-ACTING ONE-CYLINDER ENGINE

In this engine-type there occurs one pressure-stroke every fourth stroke of the piston. The cycle commences with the suction-stroke and the piston moving away from the valve-end of the cylinder. The next stroke becomes the compression-stroke, when the piston moves toward the valve-end. The third stroke is the expansion- or pressure-stroke, the piston moving away from the valve-end, and the fourth stroke is the exhaust- or discharge-stroke, when the piston again moves toward the valve-end of the cylinder. Each stroke corresponds to a course of 180 degrees by the swing of the crank; the whole cycle, therefore, occupying 4 times 180 degrees or 720 degrees.

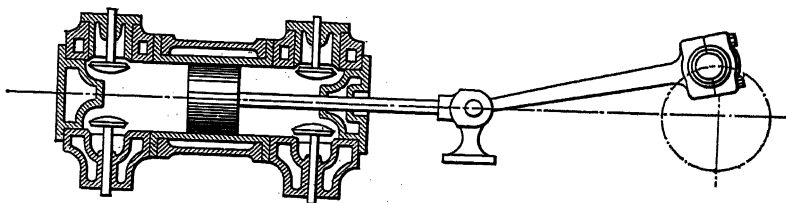
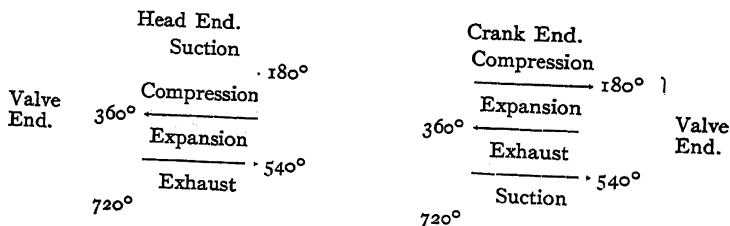
The cycle is conveniently represented in a scheme as the following:

Suction
0° ——————→ 180°
Compression
Valve End. 360° ←——————
Expansion
——————→ 540°
Exhaust
720°

TYPE II

Single-Acting Two-Cylinder Opposed Engine

The scheme for the cycle will be:

The expansion-strokes occur every second time 180 degrees and every second time 540 degrees apart.

TYPE III

Double-Acting One-Cylinder Engine

The scheme for the cycle will be:

The expansion-strokes occur every second time 180 degrees apart and every second time 540 degrees apart, the same as in Type II.

TYPE IV

Single-Acting Two-Cylinder Twin Engine

The scheme for the cycles will be:

TYPE V

Single-Acting Two-Cylinder Tandem Engine

The scheme for the cycles will be:

Rear Cylinder.

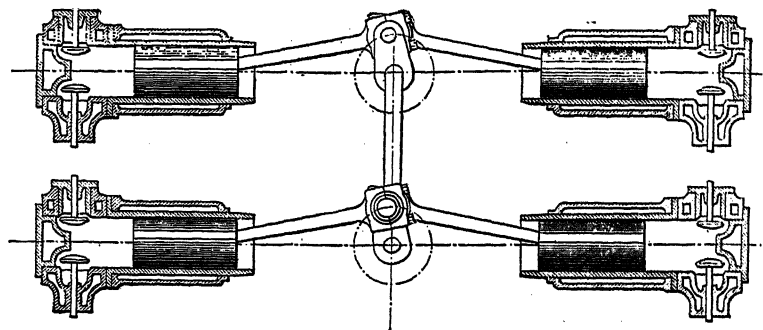
Suction
180°
Compression
Valve End. 360°
Expansion
540°
Exhaust
720°

Front Cylinder.

Expansion
180°
Exhaust
Valve End. 360°
Suction
540°
Compression
720°

The expansion-strokes of the two preceding engine-types occur uniformly 360 degrees apart.

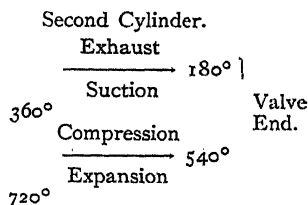
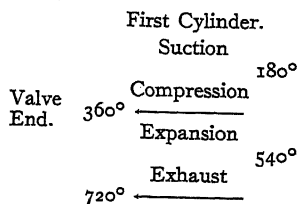
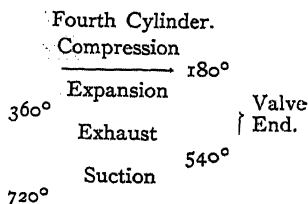
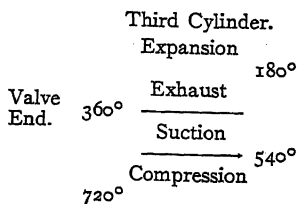
TYPE VI

Twin Single-Acting Four-Cylinder Opposed Engine

The scheme for the cycles will be:

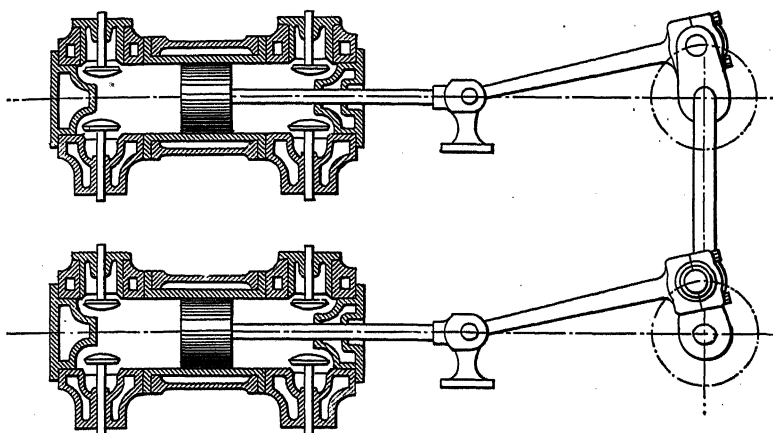
Left-hand Engine.

Right-Hand Engine.

Third Cylinder.
Expansion
180°
Valve End.
360°
Exhaust
Suction
540°
Compression
720°

Fourth Cylinder.
Compression
180°
Expansion
360°
Valve End.
Exhaust
540°
Suction
720°

The expansion-strokes occur every 180 degrees apart.

TYPE VII

Double-Acting Two-Cylinder Twin Engine

The scheme for the cycles will be:

Right-hand Cylinder.

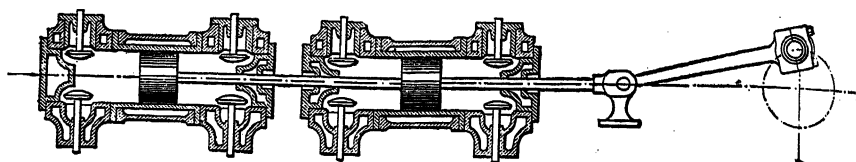
Valve End.	Head End.			Crank End.		Valve End.
	Suction	180°		Exhaust	180°	
	Compression		360°	Suction		
360°	Expansion	540°		Compression	540°	
	Exhaust			Expansion		
			720°			

Left-hand Cylinder.

Valve End.	Head End.			Crank End.		Valve End.
	Expansion	180°		Compression	180°	
	Exhaust		360°	Expansion		
360°	Suction	540°		Exhaust	540°	
	Compression			Suction		
720°			720°			

The expansion-strokes occur 180 degrees apart the same as Type VI.

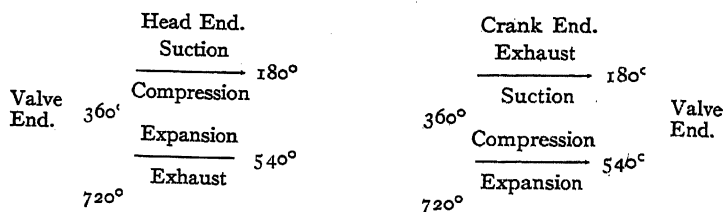
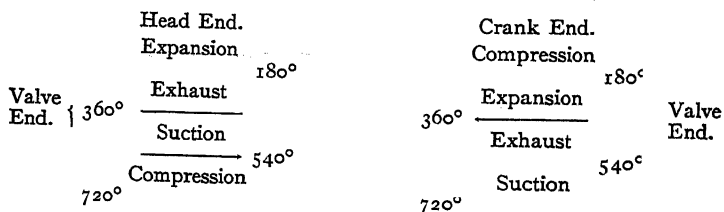
TYPE VIII

Double-Acting Two-Cylinder Tandem

The scheme for the cycles will be:

Front Cylinder.

	Head End.		Crank End.		
	Suction	180°	Exhaust	180°	
Valve End. 360°	Compression		Suction	360°	Valve End.
	Expansion	540°	Compression	540°	
720°	Exhaust		Expansion	720°	

Rear Cylinder.

	Head End.		Crank End.		
	Expansion	180°	Compression	180°	
Valve End. 360°	Exhaust		Expansion	360°	Valve End.
	Suction	540°	Exhaust	540°	
720°	Compression		Suction	720°	

The expansion-strokes occur every 180 degrees apart.

TYPE IX

Twin Double-Acting Tandem Four-Cylinder Engine

When the cranks are set 90 degrees apart, which is the most favorable for smooth running, there are obtained two sets of cycles according to the preceding scheme, and they are offset 90 degrees toward each other, giving one expansion-stroke every 90 degrees apart.

Fly-Wheel Theory.—The Tangential Crank-Effort.—The tangential crank-effort, or the force representing the driving effort at the crank-pin in a tangential direction to its orbit, varies materially for different positions of the crank. It is always zero at the beginning and at the end of each stroke, and between these points it may be in a direction so as either to accelerate or retard the forward motion of the wheel.

The variations in the tangential crank-effort for successive positions of the crank-pin may be represented, graphically, by a crank-effort curve plotted from the pressures known to exist in the cylinder for successive positions of the crank.

The Crank-Pin Pressure.—Fig. 46 is a normal indicator

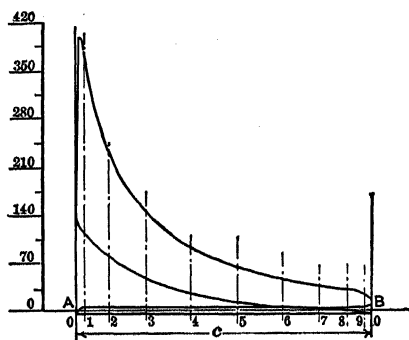



Fig. 46.

diagram representing the four-stroke cycle, on which are drawn nine ordinates perpendicularly to the atmospheric line *A B*. The distance between the atmospheric line and any of the four lines of the cycle, the suction-line, compression-line, expansion-line and exhaust-line, measured on any of the ordinates, represents the pressure on the piston at a corresponding point of the stroke. This pressure is, by means of the connecting-rod, transmitted to the crank-pin, but it is materially modified by a force evolved at the acceleration or retardation of the reciprocating parts. The latter force tends to relieve the pressure on the crank-pin at the beginning of each stroke, by subtracting from the effective pressure on the piston, and at the end of each stroke it tends to increase

the crank-pin pressure, by adding to the effective pressure on the piston.

The Accelerating Force.—The force required to accelerate (or to retard) the reciprocating parts at the head end of the piston-stroke is

$$P_1 = \pm \frac{12\,G}{g}\,\frac{V^2}{r}\left(1 + \frac{r}{l}\right),^* \quad . \quad . \quad . \qquad (100h)$$

and the force required to accelerate (or to retard) the reciprocating parts at the crank end of the piston-stroke is

$$P_2 = + 12\,G\,\frac{V^2}{r}\left(1 - \frac{r}{l}\right). \quad . \quad . \quad . \quad . \qquad (100c)$$

At the point X, Fig. 47, where the velocity of the crosshead changes from an accelerating to a retarding one, the force is

$$P_x = \pm O. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (100x)$$

The notations in these equations are:

P_1, P_2, P_x the accelerating or retarding force in pounds;
G the total weight of the reciprocating masses;
V the velocity of the crank-pin, in feet per second, $= \frac{\pi r N}{360}$;
N the number of turns of the wheel, per minute;
g the acceleration due to gravity $= 32.16$;
r the crank-radius, in inches;
l the length of the connecting-rod, in inches.

The numerical factor of the equations for the force due to the acceleration of the reciprocating parts, for the following three ratios of r to l, becomes,

$\frac{r}{l}$	$\frac{1}{5}$	$\frac{1}{5.5}$	$\frac{1}{6}$
$\frac{12}{g}\left(1 + \frac{r}{l}\right) =$	0.448	0.441	0.435
$\frac{12}{g}\left(1 - \frac{r}{l}\right) =$	0.298	0.304	0.310

* For the derivation of this expression see page 472 of the Appendix.

The ratio $\frac{r}{l}$ being seldom more than $\frac{1}{5}$ nor less than $\frac{1}{6}$, we may say, with a fair approximation for any practical case, that

$$\frac{12}{g}\left(1 + \frac{r}{l}\right) = 0.441,$$

and

$$\frac{12}{g}\left(1 - \frac{r}{l}\right) = 0.304.$$

As, in the indicator diagram, pressures are represented by pounds per square inch of the piston area, it will be convenient to express the accelerating forces also in pounds per unit area of the piston. This may be done by dividing each side of equations 100*h* and 100*c* by the area of the piston, F, whereby is obtained:

The force, per square inch piston area, required for the acceleration (or retardation) of the reciprocating parts at the head end of the stroke.

$$\frac{P_1}{F} = \pm 0.441 \frac{G}{F} \frac{V^2}{r}, \quad . \quad . \quad . \quad (100h\text{-}a)$$

and the force, per square inch piston area, required for the acceleration (or retardation) of the reciprocating parts at the crank end of the stroke.

$$\frac{P_2}{F} = \pm 0.304 \frac{G}{F} \frac{V^2}{r}. \quad . \quad . \quad . \quad (100c\text{-}a)$$

The force, per square inch area at the point X

$$\frac{P_x}{F} = \pm O, \quad . \quad . \quad . \quad . \quad . \quad (100x\text{-}a)$$

If in these equations be inserted the value of $\frac{V^2}{r}$ expressed by the number of revolutions we obtain

$$\frac{P_1}{F} = \pm 0.000034 \frac{G}{F} N^2 r, \quad . \quad . \quad . \quad (101h)$$

and

$$\frac{P_2}{F} = \pm 0.000023 \frac{G}{F} N^2 r. \quad . \quad . \quad . \quad (101c)$$

These equations, it should be noted, are correct only when $l = 5\frac{1}{2} r$, but in any practical case they are sufficiently close approximations.

Tables of Weights of the Reciprocating Parts.—The weight, G, of the reciprocating parts for an engine not being definitely known, generally, at the time the crank-effort diagram is due to be laid out, it becomes convenient to give this factor an approximate value, in accordance with previous practice with engines of types similar to the one subjected to computation.

The following tables give the usual weight of the reciprocating parts, per square inch piston area, for various classes of engines and for different cylinder sizes. From these may be selected a value for G suitable for most any case.

In the weight of the reciprocating parts there should properly be included ½ of the weight of the connecting-rod; ½ of its weight being counted as revolving. The weight, $\frac{G}{F}$, of the reciprocating parts per square inch of the piston quoted in the tables includes a proportionate part of the weight of the connecting-rod.

TABLE XIX.

For Automobile Engines

Cylinder diameter, inches:	3½	4	4½
Length of stroke, inches:	4	5	6
$\frac{G}{F}$ per each cylinder:	0.8	0.9	1.0.

TABLE XX.

For One-Cylinder Single-Acting Four-Cycle Engines

Diameter of cylinder.....	less than 6″	6″ to 12″	12″ to 18″	18″ to 24″
G	2 to 2.5	2.5 to 4	4 to 5	4.75 to 5.5

LARGE ENGINES WITH WATER-COOLED PISTONS.

For double-acting, single-cylinder four-cycle engines: use for estimate 9 pounds per square inch piston area.

For double-acting, single-cylinder two-cycle engines: use for estimate 9 pounds per square inch piston area.

For double-acting, two-cylinder, tandem engines: use for estimate 17 pounds per square inch piston area.

The Acceleration Curve for the Reciprocating Parts.—In Fig. 47 are represented the paths for the piston-pin and crank-pin, with the crank and connecting-rod shown in the positions they occupy when the connecting-rod is tangential to the path of the crank-pin. The ratio $\frac{r}{l}$ may be assumed to be 5.5. At point A, representing the beginning of the outward stroke of the piston, we offset, on the negative side of the base-line, $A\ B$, and to the same scale as that of the indicator card, Fig. 46, $A\ C = \frac{P_1}{F}$ = the accelerating force per square inch of piston area, and at the point B, representing the end of the stroke, we offset, on the positive side of the base-line, $B\ D = \frac{P_2}{F}$. At X, at which point the connecting-rod stands tangential to the crank-pin circle, the velocity changes from an accelerating to a retarding one, and the accelerating force is in that point $\frac{P_x}{F} = O$.

The distance x of point X from the perpendicular M M, representing the middle of the piston travel, changes slightly for different ratios $\frac{r}{l}$. It becomes

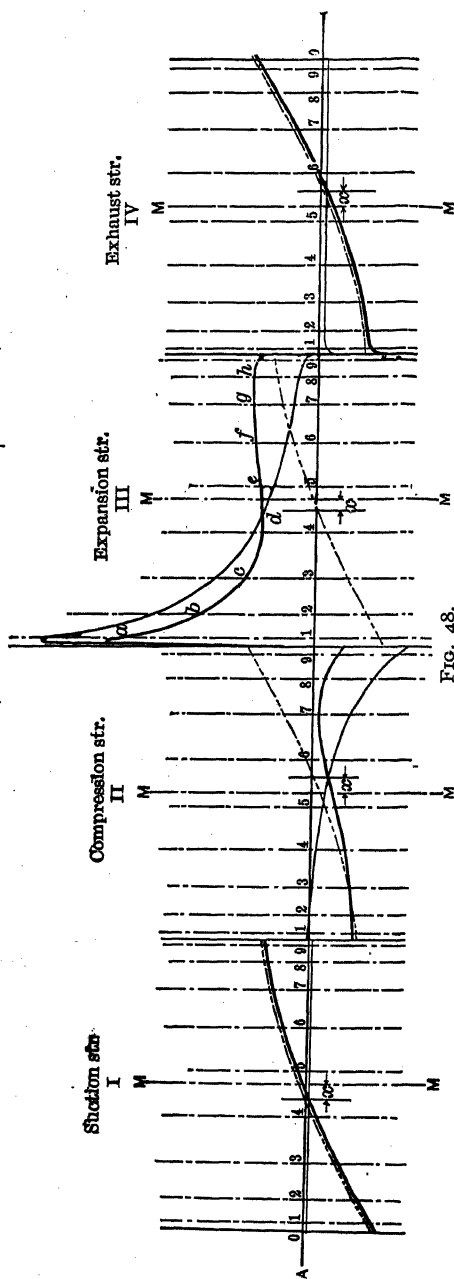
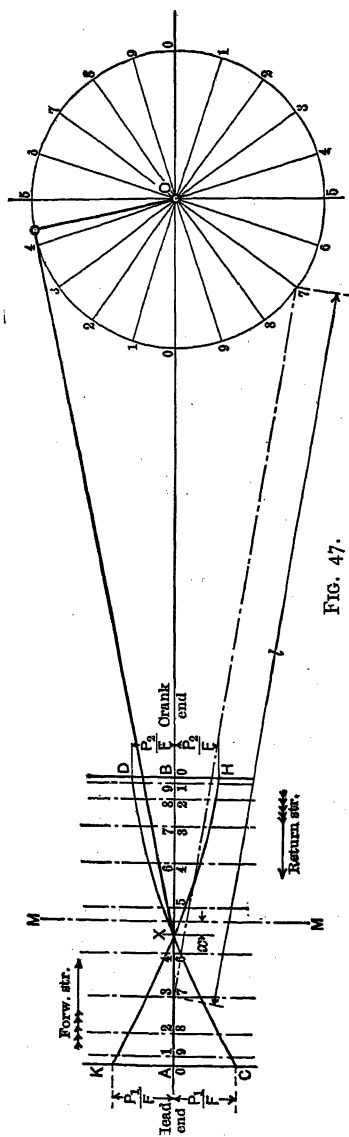
for	$\frac{r}{l}$	=	$\frac{1}{6}$	$\frac{1}{5.5}$	$\frac{1}{5}$
	x	=	$0.07\,r$	$0.08\,r$	$0.09\,r$,

but as a mean it is, x = 8 per cent of r, and for convenience this value may be used in all cases.

After locating the point X, on the base-line $A\ B$, the proper distance from M M, we draw a smooth curve through the points C X D, and obtain then the curve of force due to the acceleration of the reciprocating parts. The accelerating force for the return stroke will be negative at B and positive at A, and the curve of force H X K due to the acceleration of reciprocating parts for the return stroke will cut the base-line $A\ B$ at X.

Forw. str.
Return str.
Head end
Crank end
Suction str. I
Compression str. II
Expansion str. III
Exhaust str. IV

FIG. 47.

FIG. 48.

The curves $C X D$ and $H X K$ are parabolic, but they may, without appreciable error, be drawn as circular arcs.

The Continuous Diagram of the Horizontal Force on the Crank-Pin.—For the sake of better clearness we re-draw the normal diagram representing the pressure in the cylinder during the cycle, and give each stroke of the piston a separate space, so as to obtain a continuous pressure-diagram as shown in light lines in Fig. 48.

Each stroke is subdivided by nine ordinates, the spacing of which corresponds to nine consecutive positions of the crank-pin, spaced 18° apart, and counted from the dead centres. The manner of obtaining the spacing for the ordinates is shown in Fig. 47, and, when transporting it to the four strokes of the diagram, Fig. 48, it will be necessary to note the difference between the spacing toward the head and crank end of each stroke, and place it accordingly.

Offset in Fig. 48 the negative or positive accelerating forces due to the reciprocating parts, at the beginning and at the end of each stroke, observing that the greater positive or negative force is always plotted on the ordinate toward the head end of the stroke. Locate the cutting point X, on the base line $A B$, 0.08 r from the mid-stroke ordinates M M, toward the head end of the stroke, and draw the curves of force due to the acceleration of the reciprocating parts, as shown in the diagram in broken lines.

The diagram can conveniently be drawn in such a scale as to make the full length of it 12 inches, and to a vertical scale 100 pounds per inch.

We can now combine the continuous pressure-diagram and the curves of force due to the acceleration of the reciprocating parts, by adding, or subtracting, the pressures at the various ordinates, and obtain the curves shown in heavy lines in the diagram. The ordinates under these curves represent the resultant horizontal pressures on the crank-pin. The pressures are positive, representing a promoting force, when they appear above the base-line, and negative, or a resisting force, when they appear below the base-line.

The Tangential Crank-Pin Pressure.—To transform the result-

ant horizontal pressures on the crank-pin into tangential crank-pin pressures proceed as follows:

Draw a circle, *A O*, Fig. 49, representing the crank-circle, and divide the circumference in twenty equidistant points and number them consecutively in accordance with the numbering on the diagram, Fig. 48. Draw radial lines of proper length through points 1, 2, 3, 4, etc., and on each of these offset from the points 1, 2, 3, 4, etc., the lengths of the resultant horizontal crank-pin pressures taken from corresponding points of the diagram, Fig. 48.

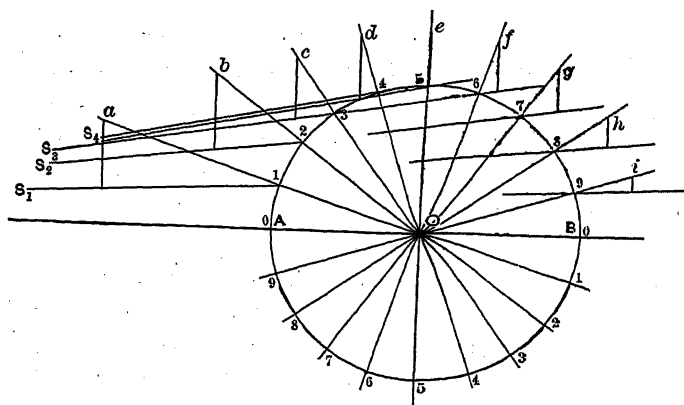



FIG. 49.

Thus we have, for instance, for stroke III of the cycle, the lengths 1 − *a*, 2 − *b*, 3 − *c*, 4 − *d*, etc., in Fig. 49 plotted to the same length as 1−*a*, 2−*b*, 3−*c*, 4−*d*, etc. Fig. 48.

Draw, further, the centre-line of the connecting-rod for each crank-position, as: 1 − s_1, 2 − s_2, 3 − s_3, etc. The lengths of the perpendiculars drawn toward *A B*, from each of the points *a*, *b*, *c*, etc., to the centre-line of the connecting-rod for the corresponding crank-position will be the tangential crank-pin pressures for the positions 1, 2, 3, etc.*

Lay off, in this manner, the length of the tangential crank-pin pressure for all points of the cycle.

The Tangential Cank-Effort Curve.—Let *A B*, Fig. 50,

* For proof of this see page 475 of the Appendix.

represent the length of the crank-pin orbit for one complete cycle. That is, for a four-stroke cycle, $A B = 2 L$ represents $4 \pi r$. Erect on $A B$ four groups of equidistant ordinates, each group containing ten ordinates representing one stroke of the cycle. Number the ordinates in degrees, or in conformity with the numbering of the crank-pin circle, Fig. 47; plot, on each, the tangential crank-pin pressure for the corresponding point, taken from Fig. 49; and through the points thus obtained draw the curve as shown.

This curve is, then, the crank-effort curve, and it shows the intensity and direction of the tangential effort for every position of the crank.

The Areas of Work Performed.—The areas below and above the base-line $A B$ represent the negative, or positive, work transmitted from the fly-wheel to the crank-pin, or from the crank-pin to the fly-wheel of the engine, and the algebraic sum of all the areas represents the total work done during the cycle. This sum should be determined—most conveniently by means of a planimeter—and let it be called A square inches.

The average value of the tangential effort for the cycle, in pounds per square inch piston area, is the ratio

$$\frac{A S_1}{2 L} = T,$$

when S_1 is the vertical scale of the diagram, in pounds per inch.

The force T we mark off on the diagram, above the base-line $A B$, as $A C$, and draw the horizontal line $C D$. The area $A B D C$ represents, then, the total work done during the cycle $= A_1$ foot-pounds per square inch piston area.

The work A_1 can be expressed by the relation

$$A_1 = 4 \pi \frac{r}{12} T,$$

or by

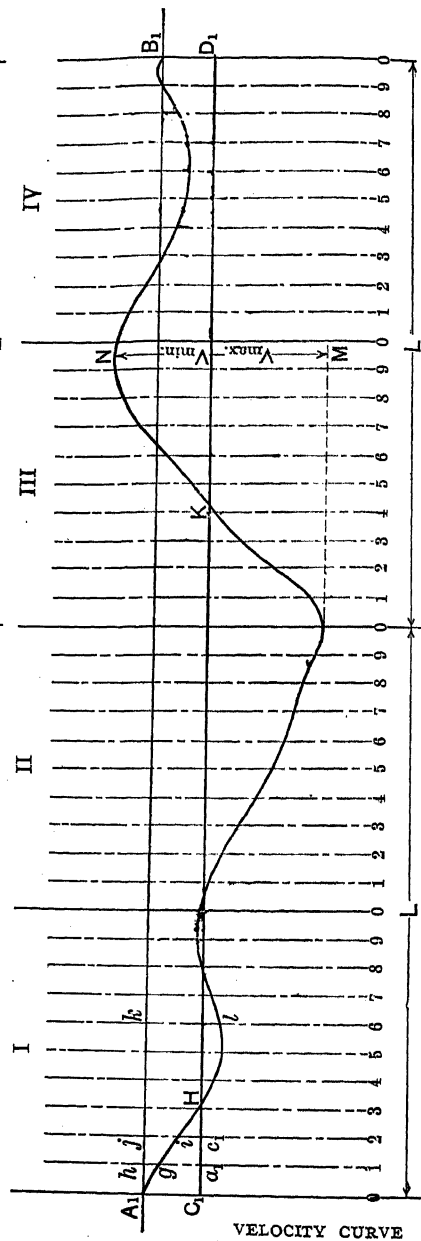
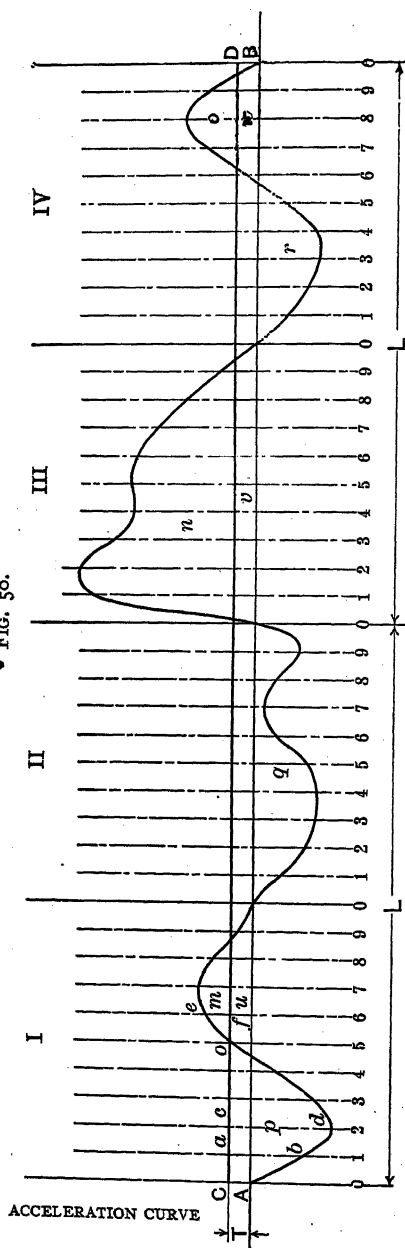
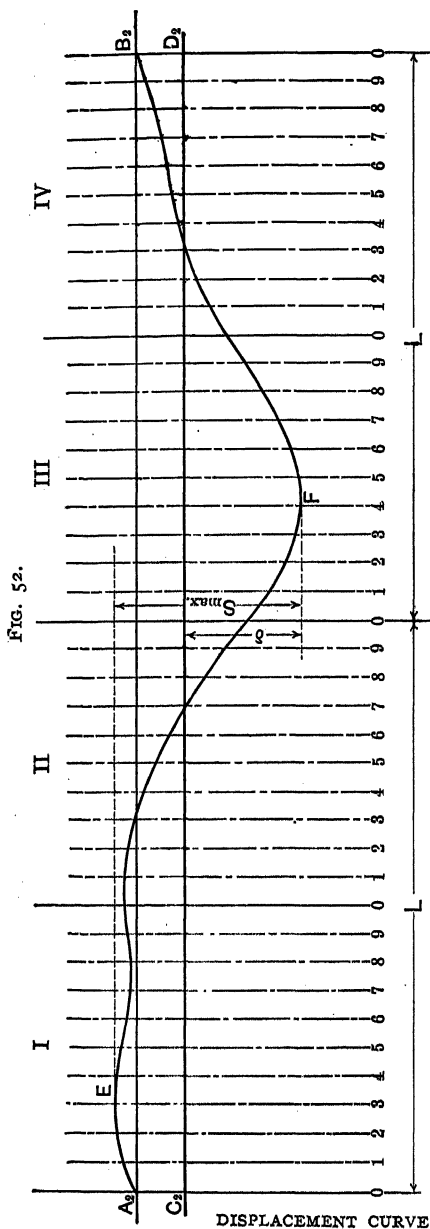
$$A_1 = 2 \frac{r}{12} p_{mc},$$

when p_{mc} is the mean effective pressure of the indicator-card.

Hence we obtain $$T = \frac{p_{mc}}{2 \pi} \text{ pounds.}$$

Fig. 50.

Fig. 51.

FIG. 52.

The height $A\,C$ of the diagram should check with this value of T.

Let each of the areas below and above the line $C\,D$ be designated by the letter it contains; the area $n + v$, thus, representing the work done during the expansion-stroke and u, v and w the narrow areas between the base-line $A\,B$ and the line $C\,D$ below the areas m, n and o. We can then write

$$(m+u)+(n+v)+(o+w)+ \\ +[A-(u+v+w)]- \\ -(p+q+r)=A.$$

Hence, $m + n + o = p + q + r$.

That is, according to the diagram, the sum of the areas above the line of normal velocity, $C\,D$, is equal to the sum of the areas below.

That this must be so is self-evident, because the same amount of work that the wheel absorbs, at high velocity, it must deliver when the velocity is below normal.

It is further evident that the largest area of accelerating work, above or below the line of normal velocity, $C\,D$, will cause the wheel to deviate the farthest from its normal speed, and that the weight of

the fly-wheel should, therefore, be determined with respect to this area.

In Fig. 50 the area n is, according to measurements, the area of maximum accelerating work, and its value we assume to be "a" square inches. If L has been made proportionately to the length c of the indicator card, and the vertical scale of the diagram the same as the vertical scale of the indicator card, thus $L = \pi c$, and $S_1 = S$, then the area a represents directly, in terms of the area of the indicator card, the maximum accelerating work in foot-pounds. But as the length L, as well as the vertical scale, may have been chosen arbitrarily, we have generally the area of the maximum accelerating work $= \frac{S_1}{S} \frac{\pi c}{L} a$.

The total work generated during one cycle, per square inch of the piston, is $p_{mc} \frac{2r}{12}$ foot-pounds, and it is represented by the area of the indicator card $= \frac{p_{mc}}{S} c$,

r being the crank-radius, in inches;
p_{mc} the mean effective pressure of the indicator card, in pounds;
S the scale of the indicator-spring; and
$\frac{p_{mc}}{S}$, thus, the mean height of the card, in inches.

If we call the quotient:

$$\frac{\text{The area of maximum accelerating work}}{\text{The area of total work per revolution}} = f$$

we find for a four-cycle single-acting one-cylinder engine

$$f = \frac{\frac{S_1}{S} \frac{\pi c}{L} a}{\frac{1}{2} \frac{p_{mc}}{S} c} = \frac{\pi \frac{a}{L}}{\frac{1}{2} \frac{p_{mc}}{S_1}}. \quad . \quad . \quad . \quad . \quad 103$$

The factor f is called the coefficient for maximum fluctuation of energy, and it is dependent to some extent on the speed of the engine and on the weight of the reciprocating parts, but for engines of similar type and speed it will be found to be in a marked degree constant.

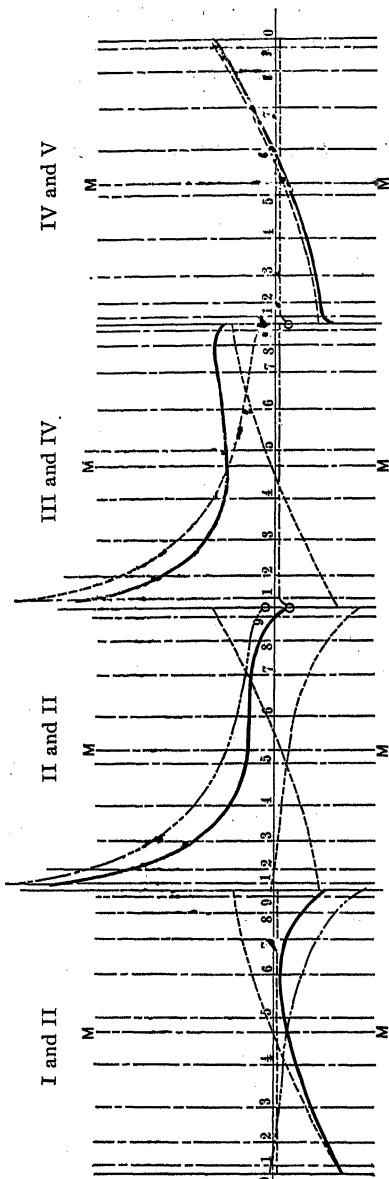



FIG. 53.

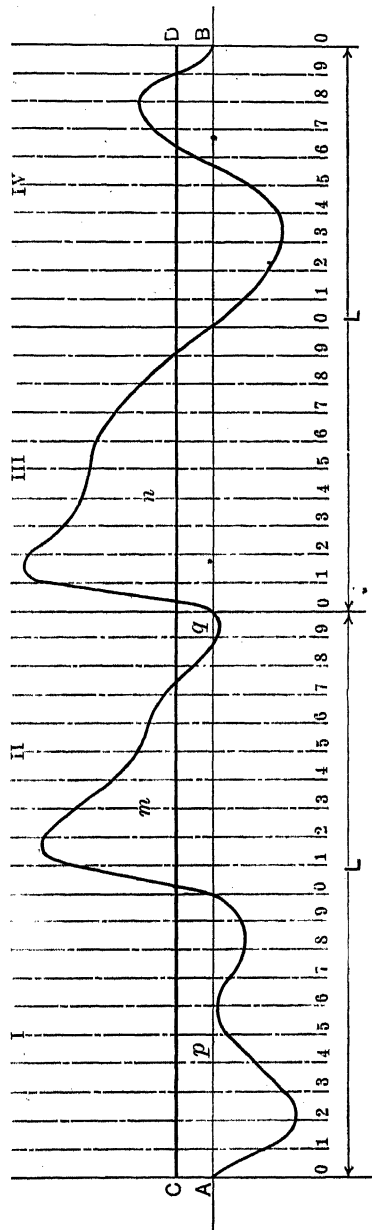



FIG. 54.

Equation 103 is correct when there is only one expansion-stroke for each two revolutions. For double-acting or multiple cylinder engines with 1, 2, or 4 expansion-strokes per revolution the value of a will change somewhat and the denominator will be increased 2-, 4- or 8-fold.

The Coefficient of Maximum Fluctuation of Energy.—Single-Acting, One-Cylinder Engine.—By measurement of the diagram, Fig. 50, we find a = 1.8 square inches, L = 4.71 inches, p_{mc} = 70 pounds, S_1 = 70 pounds per inch, and $\frac{p_{mc}}{S_1}$, thus, = 1.00.

Hence,
$$f = \frac{2 \times 3.14 \times 1.8}{4.71 \times 1.00} = 2.4.$$

Double-Acting, One-Cylinder Engine.—In the double-acting, four-cycle engine, the expansion-strokes follow each other at intervals of 180 and 540 degrees apart, measured by the swing of the crank. It may be thought, therefore, that by displacing two tangential-effort curves for a single-acting engine 180 degrees from each other and combining them we should obtain the effort-curve for the double-acting engine. This is, however, not strictly so, since the force due to the acceleration of the reciprocating parts will be different in the forward stroke from what it is for the return stroke of the piston, due to the influence of the connecting-rod. The effort-curves for the forward and return strokes become, therefore, somewhat different.

In Fig. 53 is shown the continuous diagram of the horizontal force on the crank-pin for a double-acting one-cylinder engine, and from it is obtained the crank-effort curve, Fig. 54.

It will be noticed that, while the area q detracts some from the excess velocity of the wheel, given it by the area m, only a small decrease in velocity will be effected before the area n again effects an increase. It is evident that, in this case, the area $m + n - q$ will be the maximum area of accelerating work.

By measurement we obtain:

The maximum area of accelerating work $m + n - q = a$ = 2.4 square inches, and therefore the coefficient of maximum fluctuation of energy

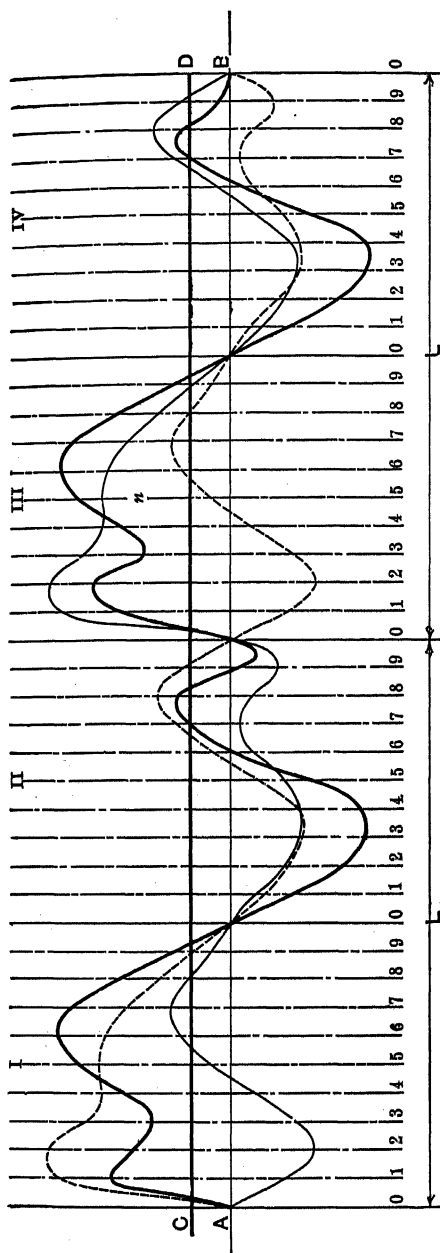



FIG. 55.

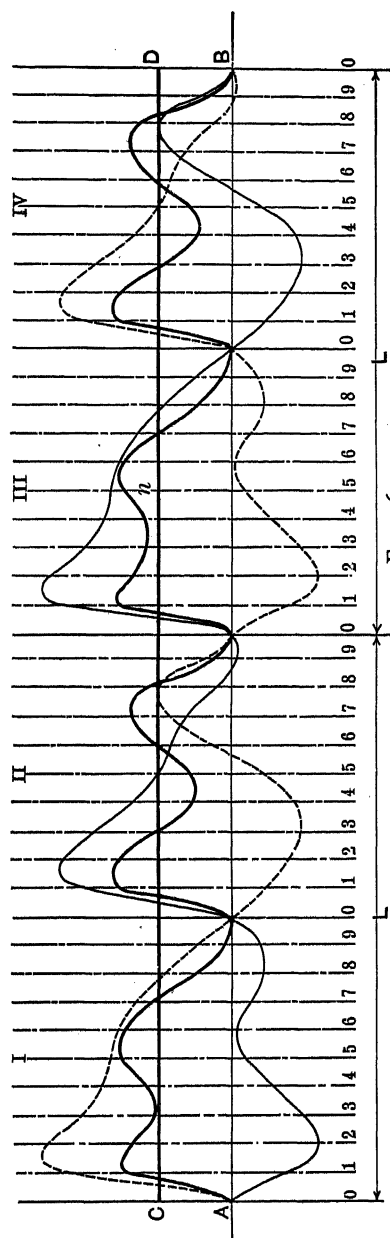



FIG. 56.

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = \frac{3.14 \times 2.4}{4.71 \times 1.00} = 1.6.$$

Single-Acting, Two-Cylinder Opposed Engine.—The reciprocating parts will in this type be of the same influence during all expansion-strokes and during similar strokes of the cycle for both cylinders. By displacing two tangential-effort curves for a single-acting engine 180 degrees, and combining them, we obtain, therefore, the effort-curve for the two-cylinder opposed engine. The resulting curve will be practically the same as that shown in Fig. 54, and the coefficient of maximum fluctuation of energy becomes

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = 1.6.$$

Single-Acting, Two-Cylinder Twin Engine.—In the two-cylinder twin engine the expansion-strokes follow each other at intervals of 360 degrees, measured by the swing of the crank. The reciprocating parts have the same influence during all expansion-strokes, and the tangential-effort curve can therefore be obtained by combining two curves of a single-acting, single-cylinder engine displaced 360 degrees from each other.

Fig. 55 represents two curves for single engines, which are displaced 360 degrees, and the combined curve is shown in heavy lines.

The area n of maximum accelerating work measures $a = 1.56$ square inches.

The coefficient of maximum fluctuation of energy becomes

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = \frac{3.14 \times 1.56}{4.71 \times 1.00} = 1.04.$$

Single-Acting, Two-Cylinder Tandem Engine.—The expansion strokes follow in this type at intervals of 360 degrees, measured

by the swing of the crank, and, as the influence due to the acceleration of the reciprocating parts is practically the same as for two single-acting single engines, the tangential-effort curve can, also in this case, be obtained by the combination of two curves for single engines, displaced 360 degrees from each other.

Fig. 55 is such a curve from which there is obtained as before: The maximum area of accelerating work $a = 1.56$ square inches. The coefficient of maximum fluctuation of energy

$$f = \frac{\pi \frac{a}{L}}{\frac{p_{mc}}{S_1}} = 1.04.$$

Twin Single-Acting, Four-Cylinder Opposed Engine.—The expansion-strokes follow each other in this type at every stroke of the engine. To construct the tangential-effort curve we combine the curves for two two-cylinder opposed engines, displaced 180 degrees, and obtain the curve Fig. 56.

The maximum area of accelerating work measures $a = 0.6$ square inches, and the work performed during the cycle is

$$4 \frac{p_{mc}}{S_1} c.$$

Hence, the coefficient of maximum fluctuation of energy becomes

$$f = \frac{\pi \frac{a}{L}}{2 \frac{p_{mc}}{S_1}} = \frac{3.14 \times 0.6}{4.71 \times 2 \times 1.00} = 0.2.$$

Double-Acting, Two-Cylinder Twin Engine.—The tangential-effort curve is obtained by combining two curves for a double-acting single-cylinder engine, as shown in Fig. 56.

The maximum area of accelerating work measures $a = 0.6$ square inches, and the work performed per revolution is $2 \frac{p_{mc}}{S_1} c.$

The coefficient of maximum fluctuation of energy, therefore,

$$f = \frac{\pi \frac{a}{L}}{2 \frac{p_{mc}}{S_1}} = \frac{3.14 \times 0.6}{4.71 \times 2 \times 1.00} = 0.2.$$

Double-Acting, Two-Cylinder Tandem Engine.—The tangential-effort curve is obtained, also for this type, by combining two curves for a double-acting single-cylinder engine, displaced 180 degrees from each other.

The coefficient of maximum fluctuation of energy becomes, as before,

$$f = \frac{\pi \frac{a}{L}}{2 \frac{p_{mc}}{S_1}} = 0.2.$$

Twin, Double-Acting Tandem Four-Cylinder Engine.—This type of engine can be arranged with the cranks set at 180 degrees or set at 90 degrees to each other. In the former case double expansion-lines follow each other at intervals of 180 degrees by the swing of the crank, in the latter case one expansion-line occurs every 90 degrees apart.

When the cranks are set at 180 degrees, the effort-curve is obtained by superposing two curves of a two-cylinder double-acting tandem engine, one over the other, and combining them. The result will be an area of maximum accelerating work twice the area for the single curve. The area of the total work will also be doubled, wherefore the coefficient of maximum fluctuation of energy becomes the same as for the two-cylinder tandem,

or
$$f = \frac{\pi \frac{a}{L}}{4 \frac{p_{mc}}{S_1}} = \frac{3.14 \times 1.2}{4.71 \times 4 \times 1.00} = 0.2.$$

When the cranks are set 90 degrees apart, the two curves of a two-cylinder double-acting tandem engine are combined after being displaced 90 degrees from each other, and the area of maximum accelerating work becomes $a = 0.4$.

Hence, the coefficient of maximum fluctuation of energy will be

$$f = \frac{\pi \frac{a}{L}}{4 \frac{p_{mc}}{S_1}} = \frac{3.14 \times 0.4}{4.71 \times 4 \times 1.00} = 0.07.$$

The deductions of the value of the coefficients f, in the preceding, have all referred to engines of the four-cycle type. The coefficients for the different types of two-cycle engines are practically the same as for the engine of the four-cycle type in which the expansion-strokes succeed each other at identically the same intervals. For instance, in a two-cycle single-cylinder engine, the expansion-strokes occur every revolution, at intervals of 360 degrees by the swing of the crank, the same as in the four-cycle single-acting twin engine. The coefficient for the maximum fluctuation of energy becomes also practically the same as for the latter type, viz.:

$$f = 1.04.$$

It will be evident that the coefficient varies somewhat for different weights of the reciprocating parts, and with the speed of the engine. Its average values for various engine-types are given in the second column of Table XXII, pages 220 and 221.

The Weight of the Fly-Wheel.—By the tangential-effort curves constructed in the preceding, there has been established, for various engine-types, the maximum value of the work generated, at one time or other during the cycle, in excess of that immediately absorbed by the normal resistance. This work, which tends to accelerate the speed of the engine from its minimum to its maximum, is for a four-cycle engine:

$$\text{maximum accelerating work} = 33{,}000 \, f \, \frac{I.H.P.}{N} \text{ foot-pounds;}$$

$\frac{I.H.P.}{N}$ being the total work generated per revolution.

The function of the fly-wheel is to absorb the accelerating work, without undue speed-variation. At a change in velocity from V minimum to V maximum, a wheel of the rim-weight W pounds absorbs the work

$$\frac{W}{2g}(V_{max.}^2 - V_{min.}^2) \text{ foot-pounds.}$$

Hence the equation for the maximum change in rim-velocity is

$$\frac{W}{2g}(V_{max.}^2 - V_{min.}^2) = 33{,}000 f, \frac{I.H.P.}{N}, \quad . \quad (104)$$

when

$I.H.P.$ designates the indicated horse-power generated;
N the number of revolutions per minute;
W the weight of the fly-wheel rim, in pounds, reduced to the mean radius;
R the mean radius of the wheel, in feet;
g the acceleration due to gravity;
$V_{max.} - V_{min.}$ the total change in rim-velocity.

If c feet per minute is the mean velocity of the wheel we have, approximately,

$$c = \frac{V_{max.} + V_{min.}}{2},$$

or we may write

$$c = \frac{2\pi R N}{60},$$

thus

$$(V_{max.} + V_{min.}) = \frac{4\pi R N}{60}.$$

If this value $(V_{max.} + V_{min.})$ be inserted in the main equation, which can be written

$$(V_{max} - V_{min.})\ (V_{max.} + V_{min.}) = 2 \times 33{,}000\ g \frac{f\,I.H.P.}{N\,W},$$

we obtain

$$V_{max.} - V_{min.} = \frac{33{,}000\ g\ 60}{2\pi} \frac{f\,I.H.P.}{R\,N^2\,W}, \quad . \quad (105)$$

and

$$\frac{V_{max.} - V_{min.}}{c} = \frac{33{,}000\ g\ 60^{-2}}{4\pi^2} \frac{f\,I.H.P.}{R^2\,N^3\,W}.$$

The quantity $\frac{V_{max.} - V_{min.}}{c}$, is an expression for the steadiness in the velocity of the wheel, which may be made anything desired. Let this factor be designated by the coefficient $\frac{1}{K}$, and let the numerical factor $\frac{33{,}000\ g\ 60^{-2}}{4\pi^2}$ be substituted by its approximate value, 96,400,000.

The formula, becomes, then

$$W = K \frac{96{,}400{,}000 \, f \, I.H.P.}{R^2 N^3} \qquad (106)$$

For single-acting, one-cylinder engines the previous formula may be made more convenient by fixing in advance on an allowable rim speed S. $R^2 N^2$ becomes then $= \frac{S^2}{40}$ and the formula may be written

$$W \quad \frac{C \, I.H.P.}{N} \qquad (106a)$$

$I.H.P.$ being the indicated power of the engine,
N the number of revolutions per minute, and
C a coefficient varying with the rim-speed.

Value of the Coefficient C.

Rim Speed (of C. of Gravity of Rim).	3500 Ft.			4000 Ft.			4500 Ft.			5000 Ft.		
	Sta rd W l.	edium H Wheel	For Electric Light Engine	Stand Whe	lium H Wheel.	For Elec Light Eng	Standard Wheel.	Medium H Wheel.	Electri Engin	tandard Wheel.	H eel.	Electri Engin
K	30	50	70	30		70	30	50	70	30	50	70
	22700	27800	53000	17300	29000	40000	13700	22000	32000		18000	26000

Table XXI, page 218, gives values for K suitable for various services.

If the rim speed be figured at the periphery of the wheel, and assuming the outside radius of the wheel, R_0, to be 1.1 of its mean radius, R, then the coefficient, C, becomes approximately 20 per cent greater than the figure given in the table.

The Acceleration Curve for the Revolving Weights.—The driving effort acting tangentially on the crank-pin is represented graphically for all positions of the crank by the crank-effort curve,

Fig. 50, but by multiplying this force, measured at any point of the cycle, by the quantity $\frac{g}{W}$ we obtain the acceleration which the force gives a weight *W*. The curve, Fig. 50, can, therefore, be said also to be the acceleration curve for the revolving weight *W*.

The Velocity Curve.—Let it be assumed that the distance *A B*, Fig. 50, instead of representing the length of the crank-pin orbit for two turns of the crank, represents the time it takes the wheel to make two revolutions. This assumption we can make, because the difference between the intervals of time and the intervals of space is so small that it would hardly be measurable in the diagram.

The distance between each two ordinates of the diagram represents, then, the time for $\frac{1}{20}$ of one turn, and the length of the mean ordinate between the base-line and the acceleration curve during the interval represents the average of the variable acceleration a weight *W* is given during the elementary time $\frac{1}{20}\,\frac{60}{N}$; $\frac{60}{N}$ being the time for one revolution.

The increase in velocity of a moving object during a time-element is the product of its acceleration and the time-element;

$$\text{Velocity} = \text{Acceleration} \times \text{Time}.$$

Hence we see, that the area of each figure, such as *b a c d*, enclosed by the base-line *C D*, the acceleration curve and two ordinates represents the increase in velocity attained during the time represented by the distance between the ordinates.

In order to represent graphically the increase or decrease in velocity of the revolving masses during the cycle, we may, therefore, integrate the elementary velocity-areas such as *A C a b*, *b a c d*, etc., from the time of the beginning of the cycle at *A* until its end at *B*, plot their sum on corresponding ordinates with line $A_1 B_1$, Fig. 51, as base, and draw a curve through the points thus obtained.

For instance, the line *g h* is plotted of a length so as to represent the area *A C a b* and the length *i j* to represent the area *A C c d*,

and so forth. The increase in velocity is positive when the velocity-area is located above the base-line $C\,D$ and negative when located below. Hence, the length $k\,l$ represents the area $C\,A\,d\,O - O\,e\,f$; the area $C\,A\,d\,O$ being negative, and the area $O\,e\,f$ positive.

The vertical distance between each minimum point and the next maximum point of the curve is the measure of the increase in velocity which the revolving masses have attained during the corresponding time, and if n is the area of maximum accelerating work, the distance $N\,M$ must be the maximum change in velocity that has occurred during the cycle $= V_{max.} - V_{min.}$

The Displacement Curve.—The line $C_1\,D_1$ is the line of normal velocity, drawn so as to make the sum of the areas above the line of the same value as the areas below.

The areas, such as $g\,i\,c_1\,a_1$, between the line $C_1\,D_1$, the velocity-curve and any two ordinates are the product of the variable excess or deficiency in velocity and the time-element; and as space also is a product of velocity and a time-element,

$$\text{Space} = \text{Velocity} \times \text{Time},$$

we may integrate the areas between the velocity-curve and the line of normal velocity $C_1\,D_1$, in the same manner in which the velocity-areas were integrated, plot their sum on corresponding ordinates with $A_2\,B_2$ as base, and obtain the displacement curve $A_2\,E\,F\,B_2$, Fig. 52.

The ordinates of this curve represent the distance which a point on the fly-wheel, or any point of the revolving system, is, at any time, ahead or behind the position it would at the time have gained with a perfectly uniform velocity.

The point F, following a period of low velocity, is a minimum point—a generator pole in the revolving system there being a maximum distance behind, and E, following a period of high velocity, is a maximum point, where the pole is a maximum distance ahead of its position due to uniform velocity.

The Weight of the Fly-Wheel for a Limited Pole-Displacement.—The areas $K\,N\,B_1\,D_1 + A_1\,H\,C_1$, Fig. 51, represent the total displacement of a pole, counted from the minimum-point to

the following maximum-point; and in the diagram, which is a curve for a single-cylinder engine, the sum of these areas measures

$$\frac{1}{12}(V_{max.} - V_{min.}) \times A_1 B_1.$$

Hence, as $A_1 B_1$ represents $\frac{2 \times 60}{N}$ seconds, we have

$$S_{max.} = \frac{1}{12}(V_{max.} - V_{min.}) \frac{120}{N} \text{ feet.}$$

It may be assumed, referring to Fig. 52, that, on an average,

$$\delta = \tfrac{3}{5} S_{max.}$$

That is, we may assume that the fly-wheel deviates three-fifths of the total distance $S_{max.}$ toward one side of the position of normal speed, and two-fifths toward the other.

The maximum deviation of a pole from the position due to uniform speed we get

$$\delta = \frac{6}{N}(V_{max.} - V_{min.}). \quad . \quad . \quad . \quad (107)$$

This value combined with equation 105 gives for a single-acting single-cylinder engine

$$W = 10{,}000{,}000 \frac{6.054 f \; I.H.P.}{R N^3 \delta}. \quad . \quad . \quad (108)$$

For multiple-cylinder and double-acting engines the numerical factor, 6, in the preceding equation 107, will generally be a somewhat smaller value, but it may be assumed that, under the most unfavorable conditions in each case, δ varies proportionately with $V_{max.} - V_{min.}$, and the approximation expressed by equation 107 will then apply in all cases.

If the value $6.054 f$ be designated by a new coefficient, F_1, we obtain the general equation for all engine types:

$$W = 10{,}000{,}000 \frac{F_1 \; I.H.P.}{R N^3 \delta}. \quad . \quad . \quad . \quad (108a)$$

As $$\delta = \frac{2 \pi R \gamma}{360}$$

when γ is the angle of deviation of a point in the revolving system,

therefore, $$W = 10{,}000{,}000 \frac{F_2 \; I.H.P.}{R^2 N^3 \gamma}. \quad . \quad . \quad . \quad (108b)$$

If ϵ is the fluctuation in degrees phase, and P the number of poles in the generator, we have

$$\gamma = \frac{2}{P}\epsilon,$$

and, hence, also $$W = 10{,}000{,}000 \frac{F_3 \quad I.H.P.}{R^2 \; N^3} \; \frac{P}{\epsilon}. \quad . \quad (108c)$$

The values of the coefficients F_1, F_2, and F_3 are:

$$F_1 = 6.054f, \quad F_2 = \frac{360}{2\pi} 6.054f, \quad \text{and } F_3 = \frac{360}{4\pi} 6.054f;$$

and their numerical values for different engine types, expressed in round numbers, are given in Table XXII.

When the number of poles in the generator is small, equation 106 becomes of higher value than equation 108c, and the wheel should then be determined with respect to its fluctuation in velocity.

By equating the formulas 106 and 108c we obtain $P = 0.0555\ \epsilon K$;

and hence, when $\epsilon = 2.5^\circ$ and $K = 100$, then $P = 13.87$;
and when $\epsilon = 2.5^\circ$ and $K = 150$, then $P = 20.80$.

The minimum number of poles for which, accordingly, the pole-displacement should be made the basis for the computation of the fly-wheel weight required for parallel operation of alternators is:

14 poles, when K is required to be 100,
and 20 poles, when K is required to be 150;
$\epsilon = 2.5^\circ$ being assumed to be a minimum value used in practice.

The Fly-Wheel Formulas.—The weight of the fly-wheel should be determined with respect to the degree of uniformity of rotation that will be required for the special service for which an engine is intended.

As the necessity for extreme steadiness of rotation and heavy wheels varies materially for different services, and as the type of the engine has a particular influence on the steadiness a given wheel will impart to the speed, the formula by which the wheel is computed should, properly, include one factor varying with the

engine-type and one factor varying with the degree of steadiness called for.

A general formula filling these requirements, and which is correct within a very slight approximation, reads:

$$W = \frac{C f \; I.H.P. \; K}{R^2 N^3}; \quad . \quad . \quad . \quad . \quad (106)$$

C is a constant $= \frac{60^{-2} \times g \times 33{,}000}{4 \pi^2} = 96{,}400{,}000$, approximately;

g being the acceleration due to gravity.

W is the weight of the wheel reduced to the radius R;

$I.H.P.$ the maximum indicated horse-power of the engine,

R the mean radius of the wheel rim, in feet,

N the number of turns per minute,

f the coefficient of maximum fluctuation of energy

$$= \frac{\text{Maximum accelerating energy}}{\text{Total energy developed during one revolution}},$$

K is a coefficient for the allowable fluctuation in speed, expressed by the ratio of the normal speed to the maximum speed-variation;

thus $$\frac{1}{K} = \frac{\text{maximum speed} - \text{minimum speed}}{\text{normal speed}}.$$

The values of K and f may conveniently be selected from Tables XXI and XXII to suit the requirements in any special case.

TABLE XXI.

Values of Coefficient K Suitable for Various Services.

	K
For ordinary industrial purposes, belt drive	25– 35
For small electric light installations, direct current, belt-driven	50– 60
For pumping machinery direct connected to engine	60–100
For large electric light installations, direct current, belt-driven	60– 80
For large electric light installations, direct current, direct connected	90–120
For gear-wheel transmissions	90–120
For blast-engines	90–150

By formula 106 the fly-wheel is determined only with respect to its fluctuations in angular velocity. The necessity that alternating current generators, working in parallel, should rotate with

as nearly perfect synchronism as possible demands, however, a regulation of the engine, not based on speed fluctuations, but based on the maximum pole-displacement that may be allowed. By pole-displacement is meant the deviation of a pole, due to variable speed, in advance and in retard of a point revolving with it with the same mean, but perfectly uniform, speed.

The following formula determines the required weight of the revolving masses, in order to keep the angular deviations of the generator-poles inside a predetermined limit γ:

$$W = 10{,}000{,}000 \frac{F_2 \; I.H.P.}{R^2 N^3 \gamma} \quad . \quad . \quad . \quad (108b)$$

Builders of alternating-current generators specify, generally, that the uniformity of operation must be such as to limit the deviation of a pole, on either side of the position of absolute uniform speed, to a certain given number of degrees phase, often $2\frac{1}{2}°$ to $3°$.

The distance between two poles of similar polarity being counted 360 degrees phase, or the distance between two consecutive poles 180 degrees, the relation between the angular degrees, measured on the pitch-circle of the generator poles, and the number of degrees phase will be the following:

$$\gamma = \frac{2}{P} \epsilon,$$

γ being the angular degrees,
ϵ the number degrees phase, and
P the number of poles.

Formula 108*b* can, therefore, for convenience be written

$$W = 10{,}000{,}000 \frac{F_3 \; I.H.P. \; P}{R^2 N^3 \epsilon}. \quad . \quad . \quad . \quad (108c)$$

Sometimes the number of poles in a generator is not specified directly, but, instead, the number of cycles and number of revolutions are given. The number of poles can then be ascertained by the formula,

$$P = \frac{120 \times \text{number of cycles}}{\text{number of revolutions}}.$$

TABLE XXII.

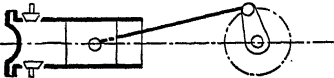
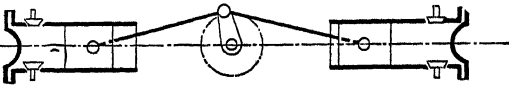
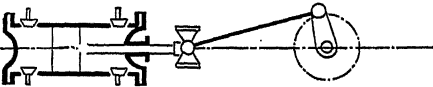
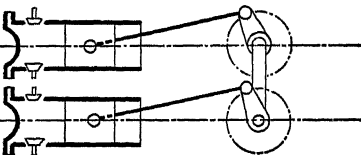
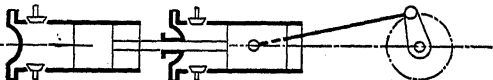
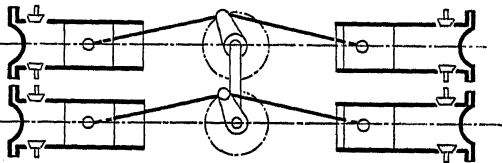
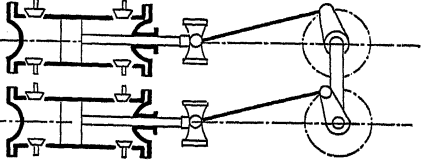
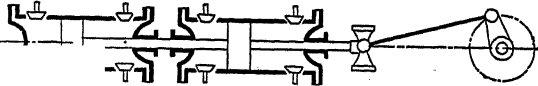
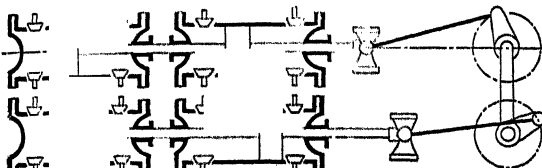
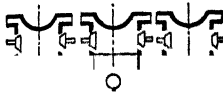
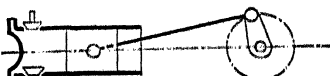
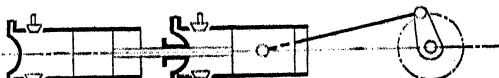
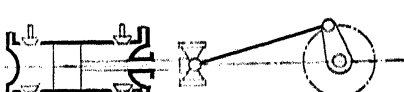
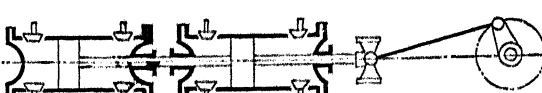
Values of Coefficients f, F_1, F_2 and F_3 for Various Engine Types.

Cylinder Arrangement.	Type.	f	F_1		F_3
Four-Cycle Engines.					
		2.4	14	800	400
	II	1.6		560	280
	III	1.6		560	280
	IV	1.04	6.4	360	180
	V	1.04	6.4	360	180
	VI			70	35
	VII			70	35
	VIII			70	35

TABLE XXII—*Continued.*

Values of Coefficients f, F_1, F_2 and F_3 for Various Engine Types.

Cylinder Arrangement.	Type.	f	F_1	F_2	
Four-Cycle Engines.					
	IX	0.07	0.44	24	
	X		3.6		105
Four-cylinder engine, same type as above; cranks 90 degrees apart.	XI			70	35
Two-Cycle Engines.					
	XII	1.04	6.4	360	180
	XIII	1.04	6.4	360	180
	XIV	0.3	1.8	104	52
	XV	0.07	0.44	24	

Example.—What diameter, and weight, of the wheel would be suitable for an 18 × 28 four-cycle, single-cylinder engine running at 190 turns per minute on suction producer-gas, the engine to be belted to a direct-current generator?

The engine will develop 90 B.H.P. on suction producer-gas, corresponding to 106 I.H.P.

The rim speed we could allow at 5,400 feet per minute, giving a wheel-diameter of 9 feet.

The required width of the belt will be, according to formula 128, in the appendix, $W = \frac{9\ B.H.P.}{V\ 0.9} = \frac{9 \times 90}{90 \times 0.9} = 10$ inches; assuming the belt to run over a small driven pulley that gives an arc of contact of about 150°.

In order to provide a wheel of ample width, including space for a set of ratchet teeth that will be required alongside one edge, the total width of the wheel-face may finally be approximately 18 inches. If the thickness of the rim be made 10 inches, the mean radius of the wheel-rim becomes $R = 4' - 1'' = 4.09$ feet.

The coefficient K, of equation 106, would in this case, according to Table XXI, be selected between the value $K = 60$ and $K = 80$, and we choose the middle value $K = 70$.

Inserting, then, the various numerical values in formula 106 we get

$$W = \frac{96{,}400{,}000 \times 2.4 \times 106 \times 70}{16.7 \times 6{,}860{,}000}$$

$$W = 15{,}000 \text{ pounds.}$$

For a twin engine of the same size cylinders as the above engine the required weight of the wheel, to give the same steadiness, would be

$$W = \frac{1.04 \times 2}{2.4}\ 15{,}000$$

$$W = 13{,}000 \text{ pounds.}$$

The proper weight of the fly-wheel for single-acting single-cylinder engines of the above dimensions, if intended for the operation of alternators in parallel, may be obtained by equation 108*c*.

Assuming the number of poles in the generator to be 14, and the maximum pole-displacement required to be inside 2½ degrees phase,

$$\text{We have}\quad W = 10{,}000{,}000 \frac{F_3 \; I.H.P.}{R^2 \; N^3} \frac{P}{\epsilon},$$

$$\text{hence,}\quad W = 10{,}000{,}000 \frac{400 \times 106}{16.7 \times 6{,}860{,}000} \frac{14}{2.5},$$

$$\text{or,}\quad W = 21{,}000 \text{ pounds, approximately.}$$

The same result will, in the case of 14 poles, be obtained by equation 106, using a coefficient $K = 100$.

The above weight includes the effective fly-wheel weight of the generator-armature; that is, its weight reduced to the radius R.

In case the number of poles were 56 instead of 14, the required total fly-wheel weight, reduced to the radius R, becomes 83,000 pounds.

A single-acting, two-cylinder opposed engine, or a double-acting, one-cylinder engine, types II and III, would require, for 56 poles, a fly-wheel weight approximately 60,000 pounds, and a single-acting twin or tandem engine, types IV and V, requires only 37,000 pounds.

It is evident, therefore, that for reducing the fly-wheel weight, when there are rigid requirements in respect to pole-displacement, the engine types IV and V give far the better results than the types I, II or III.

EXAMPLE.—As an illustration from an actual case may be quoted the engines of the A. B. Dick Co., of Chicago, Ill., which furnish electric current for the lighting of the building and for general motive power in the factory. The engines are: one single unit 14¾ × 24, rated at 55 B.H.P., and one twin engine of the same cylinder dimensions, rated at 110 B.H.P. The maximum indicated horse-power is, on producer gas, respectively, 70 and 140. Both units are direct connected to direct-current generators, and run 200 revolutions per minute.

The fly-wheels are 8′ – 6″ outside diameter and their mean radius of the rim is 3.9 feet.

The weight of the wheel for the twin-engine is	11,500 lbs.
Its weight reduced to the mean rad. of the rim	9,900 lbs.
The armature wgt. red'd. to the mean rad. of the rim	1,600 lbs.
Total wgt. red'd. to the mean rad. of the rim	11,500 lbs.
The corresponding weight for the single engine is	12,100 lbs.

Inserting the various values in formula 106, using, in the case of the twin engine, $f = 1.04$ and, in the case of the single engine, $f = 2.4$, and solving for K we obtain:

In the case of the twin engine, $K = 100$,
and in the case of the single engine $K = 90$.

The current furnished by these engines produces, under all conditions, a very satisfactory light, both when the engines run singly or in parallel. The values of the coefficient K, from 90 to 120, as quoted in Table XXI, for services such as the above, can, therefore, be considered fully conservative.

EXAMPLE.—The installation of three "Snow" four-cycle double-acting, twin-tandem engines at the Gas and Electric Light Co.'s station in San Francisco, which operate alternating-current generators very successfully in parallel, may be used as an illustration for the computation of heavy wheels.

The engines are of the following general specifications:

Four double-acting cylinders, 42 × 60. Power, 4,000 rated B.H.P., or 5,400 maximum I.H.P. The fly-wheel, 23 feet outside diameter; mean radius 11 feet. Its weight, 97,000 pounds, and it makes 88 revolutions per minute.

The generators are 25 cycle—34 poles.

For the above case we use equation 108c.

$$W = 10,000,000 \frac{F_3 \; I.H.P.}{R^2 \; N^3} \frac{P}{\epsilon}.$$

Inserting the various values, of which we obtain from Table XXII, for a twin-tandem, double-acting engine, $F_3 = 12$, and assuming that the allowed maximum fluctuation in degrees phase is $2\frac{1}{2}°$, we get

$$W = 10,000,000 \frac{12}{121} \frac{5,400}{680,000} \frac{34}{2.5} = 106,000 \text{ pounds}.$$

The actual weight of the wheel is	97,000 lbs.
Its weight reduced to the mean rad. of the rim	80,000 lbs.
The weight of the generator armature, reduced to the mean rad. of the rim, approximately	25,000 lbs.
Total weight reduced to the mean rad. of the rim, approximately, ...	105,000 lbs.

Had the number of poles of the generator been 20, instead of 34, the required weight of the wheel would become

$$W = 64{,}000 \text{ pounds.}$$

Approximately the same weight is obtained by equation 106 if the value $K = 150$ is used.

Heavy wheels, such as referred to in the above example, are in practice generally determined with guidance from displacement curves constructed as explained in the preceding, but it becomes quite possible to expedite preliminary determinations by the use of formula 108*c*, which, with the coefficients quoted in Table XXII, gives acceptable results.

EXAMPLE.—For the sake of one more illustration from practice, a determination may be made of the weight of wheels required for 44 × 54 double-acting twin tandem engines operating on blast-furnace gas, and direct-connected to 36-pole alternating-current generators. The alternators to operate in parallel at a speed of 83⅓ revolutions per minute, giving, consequently, 25 cycles.

Assuming the gas to be of such quality as to give a mean effective pressure of 68 pounds, and that the piston-rod is 12 inches in diameter, then the maximum output of the engine will be 4,300 I.H.P.

If the coefficient of the phase-displacement at a maximum output $\epsilon = 3°$ be allowed, and assuming the mean diameter of the wheel to be 21.2 feet, then the required weight of the wheel, reduced to the mean diameter, according to equation 108*c*, will be

$$W = 10{,}000{,}000 \frac{12 \times 4{,}300 \times 36}{112 \times 578{,}000 \times 3} = 92{,}500 \text{ pounds.}$$

The above data are those determining the wheels of the Allis-

Chalmers engines operating the alternating current generators at the plant of the Indiana Steel Co., at Gary, Ind.

The wheels of these engines are 23 feet outside diameter and have a rim-section 16 inches wide at the face by 21 inches deep.

The rim of the wheel weighs	68,000 lbs.
The arms and hub, approximately	23,000 lbs.
The total weight of the wheel	91,000 lbs.
The weight of arms and hub, reduced to the mean rim-dia. is	7,000 lbs.
Total weight of one wheel, reduced to the mean rim-dia.	75,000 lbs.
The weight of the revolving generator-field, reduced to the mean wheel-dia., approximately	20,000 lbs.
Total fly-wheel wgt. reduced to the mean wheel-dia., 21.2 ft.	95,000 lbs.

The normal capacity of the alternators is 2,000 K.W., but they have a guaranteed overload capacity of 30 per cent. At normal load, which accordingly is in the neighborhood of 3,200 I.H.P., the phase-displacement will be, approximately, $\epsilon = 2\frac{1}{2}^\circ$.

CHAPTER X

THE CRANK-SHAFT

In the European gas-engine practice, the custom has been general to make the engine shaft of a centre-crank type. This construction is very convenient and safe for self-contained engines, and must be considered the standard for engines of small and medium size. When, however, engines are connected in pairs, as twin engines, carrying on the shaft between them heavy wheels and generator armatures, there will, with this construction, be required to place five or more main bearings in line for the proper support of the shaft. This arrangement is by no means desirable, and will not be safe, considering the liability of the alignment of the bearings to get out of true. A better arrangement with respect to large engines, particularly of the twin type, is to use shafts with side-cranks and only two main bearings, even though it becomes necessary to make the shaft of a considerably larger diameter.

Roughly, it may be said that a side-crank shaft will be of a diameter 50 per cent larger than one with centre-crank for the same cylinder capacity. But, on the other hand, it may be made shorter, and a saving in weight is also often effected, as well as sound material insured, by forging the shaft hollow.

The principal points to be considered with respect to the design of the crank-shaft are: its strength and rigidity with reference to bending and torsional forces, and the proper bearing surfaces of its journals.

Forces Acting on the Crank-Pin.—The maximum pressure on the piston, at the time of the explosion in the cylinder, varies from, approximately, 350 pounds per square inch in engines of low compression to 450 pounds in producer-gas engines with high compression. The former figure could safely be used as basis for the computation of the shaft for gasoline or illuminating-gas engines, whereas 450 pounds would be safer with respect to producer-gas or blast-furnace gas engines.

If the maximum pressure per square inch of piston is, p, pounds, and the area of the piston, A, square inches, the maximum total pressure on the piston becomes $P = pA$. This pressure is transmitted directly to the crank-pin, when the crank passes the centre at a slow speed.

In Fig. 57 is reproduced the tangential-effort curve for the pressure stroke of a single-cylinder engine. The highest tangential pressure occurs at the point x, approximately 30 degrees from the beginning of the explosion stroke by the swing of the crank. At a corresponding distance from the head-end of the

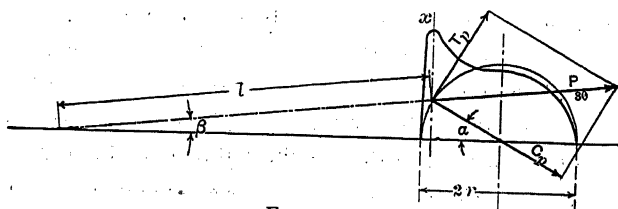



FIG. 57.

indicator-diagram the total pressure on the piston measures, approximately, 75 per cent of the total initial pressure P.

The elevation of the crank, above the centre-line of the engine, at this point being small, the pressure on the piston is practically the same as the pressure in the direction of the connecting-rod.

The pressure, P_{30}, on the crank-pin is, therefore, approximately, 75 per cent of the explosion pressure P,

$$P_{30} = 0.75\,P.$$

According to the diagram, we have

$$\sin\beta = \frac{r \sin a}{l},$$

or for the average value $\frac{r}{l} = \frac{1}{5.5}$, and $a = 30°$,

$$\sin\beta = 0.09$$

and

$$\beta = 5° - 10'.$$

Hence,

$$\cos(a + \beta) = \cos(35° - 10') = 0.817.$$

$$\sin(a + \beta) = \sin(35° - 10') = 0.576.$$

The tangential and radial forces acting on the crank-pin will, therefore be

$$T_p = sin\ (\alpha + \beta)\ P_{30} = 0.576\ P_{30}$$
$$C_p = cos\ (\alpha + \beta)\ P_{30} = 0.817\ P_{30}.$$

The strength of certain portions of the shaft is taxed most by the force, P, prevailing when the crank passes the centre, while

FIG. 58. FIG. 59.

other portions are taxed most by the force, P_{30}, prevailing when the crank stands 30 degrees from the centre. It will, therefore, be necessary to analyze the strength of the shaft with reference to both of these forces.

Small- and medium-sized engines have often the fly-wheel carried on the end of the shaft overhanging the main journal, as

FIG. 60.

Fig. 58. Large engines should always have a third bearing for the support of the end of the shaft, as Fig. 59, in order to avoid the heavy strains in the shaft and crank, due to the weight of the wheel, that otherwise would be set up.

The following two examples explaining the method generally

employed for computing the strength of the various parts of centre-crank shafts relate, the first one, to a shaft carried by two bearings and, the second, to a shaft carried by three bearings.

The Strength of Centre-Crank Shafts Supported by Two Bearings.—Fig. 60 is a preliminary sketch for a shaft intended for a 9×16 horizontal producer-gas engine to run at 230 R.P.M. and it will be required to analyze the stresses that will obtain in the various parts.

The area of a 9-inch piston is 63.6 square inches, which for a pressure of 450 pounds per square inch gives a total pressure on the piston $P = 28{,}600$, approximately. The power of the engine will be 17 B.H.P., or 20 I.H.P., and a wheel weighing 3,000 pounds and of 6 feet diameter will give a satisfactory steadiness of rotation (the coefficient of steadiness being $K = 50$).

The Crank on Centre.—In Fig. 61 are represented diagrammatically the forces acting on the shaft, when the crank, at the time of the explosion in the cylinder, passes the head-end centre.

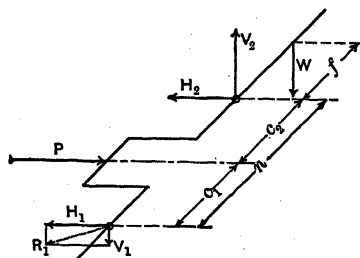
P is the total pressure on the piston, and H_1 and H_2 the reactions on the bearings due to this force. W is the weight of the wheel and V_1 and V_2 the reactions in the bearings due to it.

The symbols, c_1, c_2 and f, denote, according to the figure, the distances between the left-hand and right-hand bearings and the centre of the engine, and between the right-hand bearing and the centre of the fly-wheel; n is the distance, centre to centre, between the bearings.

Fig. 61.

The data given are:

$$c_1 = c_2 = 11''$$
$$f = 11''$$
$$n = 22''$$

$P = 28{,}600$ pounds.
$W = 3{,}000$ pounds

We obtain

$$H_1 = \frac{c_2}{n} P = \frac{11}{22} 28{,}600 = 14{,}300 \text{ pounds.}$$

$$H_2 = \frac{c_1}{n} P \quad \frac{11}{22} 28{,}600 = 14{,}300 \text{ pounds.}$$

$$V_1 = \frac{f}{n} W = \frac{11}{22} \; 3{,}000 = \; 1{,}500 \text{ pounds.}$$

$$V_2 = \frac{n+f}{n} W = \frac{33}{22} 3{,}000 = 4{,}500 \text{ pounds.}$$

$$R_1 = \sqrt{H_1^2 + V_1^2} = \sqrt{14{,}300^2 + 1{,}500^2} = 14{,}380 \text{ pounds.}$$

In Fig. 62 are represented the forces that strain the material (the same forces as in Fig. 61), and also the sections of the material offering resistance, at the various points of the shaft.

Additional data given are:

$$d = d_1 = d_2 = 5'', \quad c_1 - e = 7'',$$
$$b = 3'', \quad h = 6''.$$
$$e = 4'', \quad r = 8''.$$

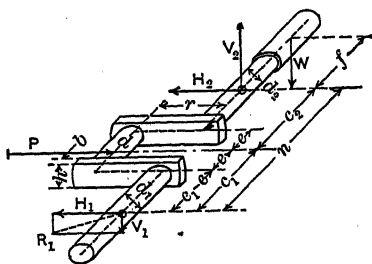



FIG. 62.

Maximum Strain at the Middle Section of Crank-Pin.- The bending moment is

$$M_b = R_1 c_1 = 14{,}380 \times 11 = 158{,}180.$$

Section modulus $J^* = 0.1\, d^3 = 0.1 \times 5^3 \quad 12.5.$

Maximum bending strain in pin

$$S_b = \frac{M_b}{J} = \frac{158{,}180}{12.5} = 12{,}650 \text{ pounds.}$$

The shearing strain at the ends of pin is

$$S_s = \frac{2\,P}{\pi\, d^2} = \frac{2 \times 28{,}600}{3.14 \times 25} \quad 733 \text{ pounds.}$$

* Section Modulus, $\frac{J}{a} = \frac{\text{Mom. of Inertia}}{d} = \frac{\pi}{32}$ appr. $0.1\, d^3$.

The twisting moment is

$$M_t = V_1 r = 1,500 \times 8 = 12,000.$$

The section modulus for torsion

$$\frac{J_t}{a}* = 0.2\, d^3 = 0.2 \times 5^{-3} = 25.$$

Maximum torsional strain,

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{12,000}{25} = 480 \text{ pounds.}$$

The latter strain is small compared with the bending strain, and can be disregarded.

Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Broad Side.—

The bending moment is

$$M_b = H_1 (c_1 - e) = 14,300 \times 7 = 100,100.$$

Section modulus

$$\frac{J}{a} = \frac{h b^2}{6} = \frac{6 \times 9}{6} = 9.$$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{100,100}{9} = 11,120 \text{ pounds.}$$

The twisting moment in a vertical section of the arm is

$$M_t = V_1 (c_1 - e) = 1,500 \times 7 = 10,500.$$

The section modulus

$$\frac{J_t}{a} = \frac{2}{9} h b^2 = \frac{2}{9} 6 \times 9 = 12.$$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{10,500}{12} = 875 \text{ pounds.}$$

* Section Modulus for Torsion, $\frac{J_t}{a} = \frac{\text{Polar Mom. of Inertia}}{}$ $\frac{\pi}{16} d^3 =$ appr. $0.2\, d^3$.

The bending and twisting strains are combined into an equivalent tensile strain by equation

$$S = 0.35\, S_b + 0.65 \sqrt{S_b^2 + 4 S_t^2}.$$

Hence,

$$S = 0.35 \times 11{,}120 + 0.65 \sqrt{\overline{11{,}120}^2 + 4 \times \overline{875}^2} = 11{,}221 \text{ pounds.}$$

The compressive strain in the arm is

$$S_c = \frac{H_1}{b\,h} = \frac{14{,}300}{18} = 794 \text{ pounds.}$$

The total maximum strain in the arm, therefore,

$$S + S_c = 12{,}015 \text{ pounds.}$$

Maximum Strain in the Right-Hand Crank-Arm, at the Face of the Broad Side.—

The bending moment is

$$M_b = H_1 (c_1 + e) - P e = 14{,}300 \times 15 - 28{,}600 \times 4 = 100{,}100.$$

This is the same value as for the left-hand crank-arm as it correctly should be.

The maximum bending strain, therefore,

$$S_b = 11{,}120 \text{ pounds.}$$

The maximum twisting moment is

$$M_t = V_1 (c_1 + e) = 1{,}500 \times 15 = 22{,}500.$$

The section modulus $\dfrac{J_t}{a} = 12.$

Maximum twisting strain

$$S_t = \frac{M_t}{\dfrac{J_t}{a}} = \frac{22{,}500}{12} = 1{,}875 \text{ pounds.}$$

The combined strain will be

$$S = 0.35 \times 11{,}120 + 0.65 \sqrt{\overline{11{,}120}^2 + 4 \times \overline{1{,}875}^2} = 11{,}516 \text{ pounds.}$$

The compressive strain in the arm is

$$S_c = 794 \text{ pounds.}$$

The total maximum strain, therefore,

$$S + S_c = 12{,}310 \text{ pounds.}$$

In case there is one wheel on each end of the shaft at the distance f from the bearings, thus $V_1 = V_2 = W$, then the torsional moment in the crank-pin becomes zero, and the bending moment due to H_1, V_1 and W becomes

$$M_1 = c_1 \sqrt{H_1^2 + \left(\frac{f}{c_1}\right)^2 V_1}$$

The heaviest strain will occur in the crank-pin and arms when the crank passes the centre. The preceding estimate is therefore sufficient, as far as the strength of the pin and arms are concerned. For comparison, however, a similar estimate for the crank-position 30 degrees from the centre will be carried out in the following:

The Crank 30 Degrees above Centre.—The crank is represented in Fig. 63 in a position 30 degrees from the head-end centre, and the forces acting on the shaft, tangentially and radially to the crank, are as denoted.

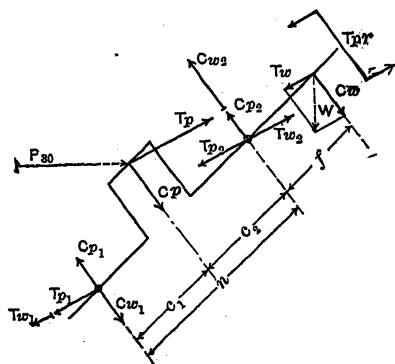
The force P_{30} is dissolved into the tangential component T_p and the radial component C_p.

The reactions in the bearings from T_p are T_{p1} and T_{p2}, and the reactions from C_p are C_{p1} and C_{p2}.

The weight of the fly-wheel is W, and T_w and C_w are the tangential and radial components due to the weight W.

FIG. 63.

The reactions in the bearings from T_w are T_{w1} and T_{w2}, and the reactions from C_w are C_{w1} and C_{w2}.

The turning resistance at the fly-wheel is represented by the moment $T_p . r$, applied at the fly-wheel hub, but the belt-pull, being small, is neglected.

The symbols c_1, c_2 and f denote, according to the figure, as before, the distances between the left-hand and right-hand bearings and the centre of the engine, and between the right-hand bearing and the centre of

the fly-wheel; n is the distance, centre to centre, between the bearings.

The data given are:

$$c_1 = c_2 = 11''.$$
$$f = 11'',$$
$$n = 22''.$$

$$P_{30} = 0.75\, P = 21{,}450 \text{ pounds}$$
$$W = 3{,}000 \text{ pounds.}$$

From these we obtain

$$C_p = \cos(\alpha + \beta)\; P_{30} = 0.817 \times 21{,}450 = 17{,}524$$
$$T_p = \sin(\alpha + \beta)\; P_{30} = 0.576 \times 21{,}450 = 12{,}356$$
$$C_{p1} = \frac{c_2}{n}\, C_p = \tfrac{1}{2} \times 17{,}524 = 8{,}762$$
$$T_{p1} = \frac{c_2}{n}\, T_p = \tfrac{1}{2} \times 12{,}356 = 6{,}178$$
$$C_{p2} = \frac{c_1}{n}\, C_p = \tfrac{1}{2} \times 17{,}524 = 8{,}762$$
$$T_{p2} = \frac{c_1}{n}\, T_p = \tfrac{1}{2} \times 12{,}356 = 6{,}178$$

$$C_w = W \sin 30^\circ = \tfrac{1}{2} \times 3{,}000 = 1{,}500$$
$$T_w = W \cos 30^\circ = 0.866 \times 3{,}000 = 2{,}600$$
$$C_{w1} = \frac{f}{n}\;\; C_w = \tfrac{1}{2} \times 1{,}500 = 750$$
$$T_{w1} = \frac{f}{n}\;\; T_w = \tfrac{1}{2} \times 2{,}600 = 1{,}300$$
$$C_{w2} = \frac{n + f}{n}\;\; C_w = \tfrac{3}{2} \times 1{,}500 = 2{,}250$$
$$T_{w2} = \frac{n + f}{n}\;\; T_w = \tfrac{3}{2} \times 2{,}600 = 3{,}900$$

$$C_{p1} - C_{w1} = 8{,}012$$
$$T_{p1} + T_{w1} = 7{,}478$$

In Fig. 64 are represented the same forces as are shown in Fig. 63, as well as the sections of the material of the shaft resisting them.

The Maximum Strain at the Middle of the Crank-Pin.—The moment acting radially

$$M_{b1} = (C_{p1} - C_{w1})\, c_1 = 8{,}012 \times 11 = 88{,}132.$$

The moment acting tangentially

$$M_{b2} = (T_{p1} + T_{w1})\, c_1 = 7{,}478 \times 11 = 82{,}258.$$

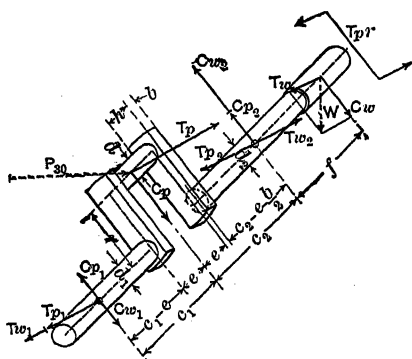



FIG. 64.

The combined moment is

$$M_b = \sqrt{M_{b1}^2 + M_{b2}^2} = \sqrt{8{,}012^{2} + 7{,}478^{2}} \times 11 = 120{,}000.$$

The section modulus is $\dfrac{J}{a} = 0.1\, d^3 = 12.5.$

The maximum bending strain in pin

$$S_b = \frac{M_b}{\dfrac{J}{a}} = \frac{120{,}000}{12.5} = 9{,}600 \text{ pounds.}$$

The maximum twisting moment is

$$M_t = (T_{p1} + T_{w1})\, r = 7{,}478 \times 8 = 59{,}824.$$

The section modulus for torsion is $\dfrac{J_t}{a} = 0.2\, d^3 = 25.$

The maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{59,824}{25} = 2,390 \text{ pounds.}$$

The combined strain thus

$$S = 0.35 \times 9,600 + 0.65 \sqrt{9,600^2 + 4 \times 2,390^2} = 10,330 \text{ pounds.}$$

The Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Broad Side.—

Maximum bending moment

$$M_b = (C_{p1} - C_{w1})(c_1 - e) = 8,012 \times 7 = 56,084.$$

The section modulus $\frac{J}{a} = \frac{h\,b^2}{6} = \frac{6 \times 9}{6} = 9.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{56,084}{9} = 6,232 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})(c_1 - e) = 7,478 \times 7 = 52,346.$$

Section modulus for torsion $\frac{J_t}{a} = \frac{2}{9}\,h\,b^2 = 12.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{52,346}{12} = 4,362 \text{ pounds.}$$

The maximum combined strain will be

$$S = 0.35 \times 6,232 + 0.65 \sqrt{6,232^2 + 4 \times 4,362^2} = 9,150 \text{ pounds.}$$

The Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Narrow Side Near Pin.—

Maximum bending moment

$$M_b = (T_{p1} + T_{w1})\left(r - \frac{d}{2}\right) = 7,478 \times 5.5 = 41,129.$$

The section modulus $\frac{J}{a} = \frac{b\,h^2}{6} = \frac{3 \times 36}{6} = 18.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{41,129}{18} = 2,285 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})\ (c_1 - e) = 52,346, \text{ as before.}$$

Section modulus for torsion $\frac{J_t}{a} = \frac{2}{9}\ b\ h^2 = 24.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{52,346}{24} = 2,181 \text{ pounds.}$$

The maximum combined strain will be

$$S = 0.35 \times 2,285 + 0.65 \sqrt{\overline{2,285}^2 + 4 \times \overline{2,181}^2} = 3,990 \text{ pounds.}$$

The Maximum Strain in the Right-Hand Crank-Arm, at the Face of the Broad Side.—

Maximum bending moment

$$M_b = (C_{p1} - C_{w1})\ (c_1 + e) - C_p\ e = 8,012 \times 15 - 17,524 \times 4 = 50,084.$$

The section modulus $\frac{J}{a} = 9.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{50,084}{9} = 5,565 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})\ (c_1 + e) - T_p\, e =$$
$$7,478 \times 15 - 12,356 \times 4 = 62,746 \text{ pounds.}$$

Section modulus for torsion $\frac{J_t}{a} = 12.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{62,746}{12} = 5,229 \text{ pounds.}$$

The maximum combined strain will be

$$0.35 \times 5{,}565 + 0.65 \sqrt{5{,}565^2 + 4 \times 5{,}229^2} = 9{,}650 \text{ pounds.}$$

The Maximum Strain in the Right-Hand Crank-Arm, at the e of the Narrow Side Near the Shaft.—
Maximum bending moment

$$M_b = T_p \left(r - \frac{d_2}{2} \right) + (T_{p1} + T_{w1}) \frac{d_2}{2}$$

$$= 12{,}356 + 5.5 + 7{,}478 \times 2.5 = 86{,}653.$$

The section modulus $\frac{J}{a} = \frac{b\,h^2}{6} = 18.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{86{,}653}{18} = 4{,}814 \text{ pounds.}$$

Maximum twisting moment

$$M_t = (T_{p1} + T_{w1})(c_1 + e) - T_p\,e = 62{,}746, \text{ as before.}$$

Section modulus for torsion $\frac{J_t}{a} = \frac{2}{9}\,b\,h^2 = 24.$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{62{,}746}{24} = 2{,}614 \text{ pounds.}$$

The maximum combined strain will be

$$= 0.35 \times 4{,}814 + 0.65 \sqrt{4{,}814^2 + 4 \times 2{,}614^2} = 6{,}304 \text{ pounds.}$$

The Maximum Strain in the Shaft, at the Middle Section of e Main Journal.—When the crank is on centre there is no isting moment at any section of the main shaft, but there ists, for any position of the crank, a bending moment *at the ntre of the journal next to the fly-wheel*, which is,

$$M_b = W\,f = 3{,}000 \times 11 = 33{,}000.$$

The section modulus for the shaft is $\frac{J}{a} = 0.1\,d^3_2 = 12.5.$

The maximum bending strain therefore

$$S = \frac{M_b}{\frac{J}{a}} = \frac{33{,}000}{12.5} = 2{,}640 \text{ pounds.}$$

When the crank stands 30 degrees from the head-end centre there is added to this a maximum twisting moment

$$M_t = T_p \times r = 12{,}356 \times 8 = 98{,}848.$$

The section modulus for torsion is $\frac{J_t}{a} = 0.2\, d^3_2 = 25.$

The maximum twisting strain

$$\frac{M_t}{\frac{J_t}{a}} = \frac{98{,}848}{25} \quad 3{,}954.$$

The combined maximum strain at the centre of the journal

$$S = 0.35 \times 2{,}640 + 0.65\sqrt{\overline{2{,}640}^2 + 4 \times \overline{3{,}954}^2} = 6{,}340 \text{ pounds.}$$

The Maximum Strain in the Section of the Main Shaft Next to the Right-Hand Crank-Arm.—

Maximum bending moment radially to the crank

$$M_{b1} = (C_{p2} + C_{w2})\left(c_2 - e - \frac{b}{2}\right) - C_w\left(f + c_2 - e - \frac{b}{2}\right)$$

$$= 11{,}012 \times 5.5 - 1{,}500 \times 16.5 = 85{,}316.$$

Maximum bending moment tangentially to the crank

$$M_{b2} \quad (T_{p2} - T_{w2})\left(c_2 - e - \frac{b}{2}\right) + T_w\left(f + c_2 - e - \frac{b}{2}\right)$$

$$= 2{,}278 \times 5.5 + 2{,}600 \times 16.5 = 55{,}429.$$

The combined bending moment

$$M_b = \sqrt{M_{b\,1}^2 + M_{b\,2}^2} = \sqrt{\overline{85{,}316}^2 + \overline{55{,}429}^2} = 101{,}740.$$

The section modulus is $\frac{J}{a} = 0.1\, d^3_2 = 12.5.$

The maximum bending strain

$$\frac{M_b}{\frac{J}{a}} = \frac{101{,}740}{12.5} \quad 8{,}140 \text{ pounds.}$$

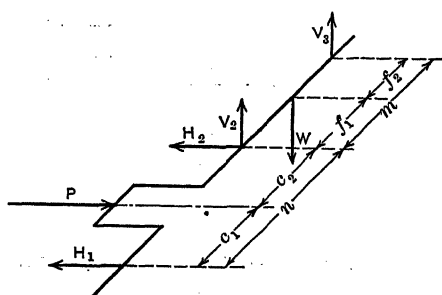
The maximum twisting moment is as before $M_t = T_p r = 98,848$, and the maximum twisting strain $S_t = 3,954$ pounds.

The combined maximum strain will be

$$S = 0.35 \times 8,140 + 0.65 \sqrt{8,140^{-2} + 4 \times 3,954^{-2}} = 10,220 \text{ pounds.}$$

The Strength of a Centre-Crank Shaft Supported on Three Bearings.—In Fig. 65 are represented all the forces acting on a shaft of this type, when the crank, at the time of the explosion, passes the centre.

FIG. 65.

P is the total pressure on the piston, H_1 and H_2 the reactions on the bearings 1 and 2 due to this force. W is the weight of the fly-wheel, and the reactions due to it in the bearings 2 and 3 are V_2 and V_3.

Assume it to be required to analyze the strains in a shaft for a 20 × 32 engine, the preliminary sketch of which is shown in Fig. 66.

Running on suction producer-gas at 160 revolutions per

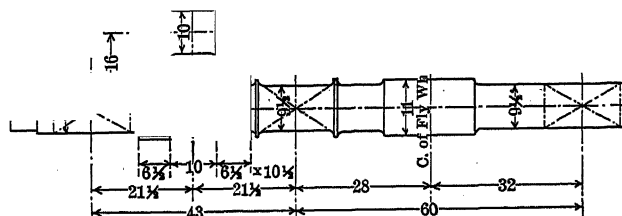



FIG. 66.

minute, the power of the engine will be approximately 110 B.H.P., or 144 I.H.P. A heavy wheel would be of a weight of 32,000 pounds, of a diameter 11′ – 0″.

The area of the piston being 314 square inches, and the maximum pressure allowed at 450 pounds per square inch, we obtain

the total maximum pressure on the piston $P = 314 \times 450 = 141,300$ pounds.

The maximum pressure on the crank when 30 degrees from the head-end centre will be $P_{30} = 0.75\ P = 106,000$ pounds.

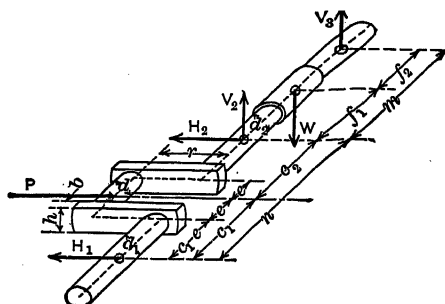



FIG. 67.

The principal dimensions of the shaft as well as the forces acting, when the crank passes the centre, are given in Fig. 67.

The data given are:

$$c_1 = c_2 = 21.5'' \qquad e = 8.25'$$
$$c_1 - e = 13.25'', \quad f_1 = 28'', \quad f_2 = 32''.$$
$$n = 43'', \quad b = 6\tfrac{1}{2}'', \quad d = 10'',$$
$$m = 60'', \quad r = 16''.$$

$$P = 141,300 \text{ pounds.}$$
$$W = 32,000 \text{ pounds.}$$

From this we obtain

$$H_1 = H_2 = \frac{c_2}{n} P = 70,650 \text{ pounds}$$

$$V_2 = \frac{f_2}{m} W \qquad \tfrac{32}{60} \times 32,000 = 17,067 \text{ pounds.}$$

$$V_3 = \frac{f_1}{m} W \qquad \tfrac{8}{0} \times 32,000 = 14,933 \text{ pounds.}$$

The Maximum Strain at the Middle of the Crank-Pin.
Maximum bending moment

$$M_b = H_1 c_1 = 70,650 \times 21.5 = 1,518,975.$$

The section modulus is $\frac{J}{a} = 0.1\ d^3 = 0.1 \times 10^{-3} \qquad 100.$

Maximum bending strain

$$S_b = \frac{M_b}{J} = 15{,}190 \text{ pounds.}$$

The Maximum Strain in the Left-Hand Crank-Arm, at the Face of the Broad Side.—

Maximum bending moment

$$M_b = H_1 (c_1 - e) = 70{,}650 \times 13.25 = 936{,}112.$$

The section modulus is $\frac{J}{a} = \frac{h\,b^2}{6} \quad \frac{10.5 \times 6.5^2}{} = 74.$

Maximum bending strain

$$S_b = \frac{M_b}{J} = \frac{936{,}112}{74} = 12{,}650 \text{ pounds.}$$

The strain due to compression

$$S_c = \frac{H_1}{b \times h} = \frac{70{,}650}{68.25} = 1{,}035 \text{ pounds.}$$

The total maximum strain in arm $S = 13{,}685$ pounds.

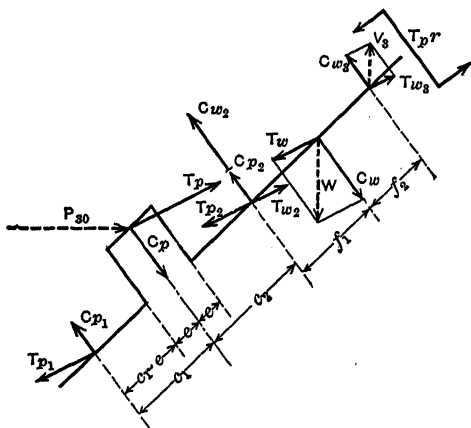



FIG. 68.

The Maximum Strains in the Right-Hand Crank-Arm, at the Face of the Broad Side are the same as above.

The strains in the shaft, at the fly-wheel hub, at the middle

of the main bearing, and in a section near the right-hand arm, should be figured with respect to the forces acting on the shaft when the crank stands 30 degrees from the head-end centre. These forces are shown diagrammatically by Fig. 68. P_{30} is the force transmitted by the connecting-rod to the crank-pin and C_p and T_p its components, radially and tangentially to the crank.

The component C_p gives the reactions C_{p1} and C_{p2} at the bearings 1 and 2 and the component T_p gives the reactions T_{p1} and T_{p2}. The fly-wheel weight is represented by W, and its components, radially and tangentially, are C_w and T_w.

The reactions in bearings 2 and 3, due to the radial component,

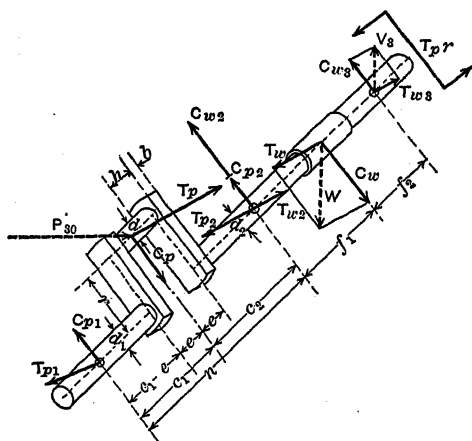



FIG. 69.

C_w, are respectively C_{w2} and C_{w3}, and the reactions due to the tangential component, T_w, are T_{w2} and T_{w3}. $T_p . r$ is a turning moment acting at the fly-wheel hub, representing the resistance to the rotation.

In Fig. 69 are represented the same forces as are shown in Fig. 68, as well as the sections of the material resisting them.

The data given are:

$c_1 = c_2 = 21.5'$ $\qquad f_1 = 28''$, $\qquad f_2 = 32,''$
$m = 60''$, $\qquad r = 16''$, $\qquad e = 8.25''$,
$d_1 \qquad d_3 = 9\frac{1}{2}''$, $\qquad d_4 = 11''$.

$$P_{30} = 0.75\ P = 106{,}000 \text{ pounds.}$$
$$W = 32{,}000 \text{ pounds.}$$

From this is obtained

$$C_p = \cos\,(\alpha + \beta)\ P_{30} = 0.817 \times 106{,}000 = 86{,}600 \text{ pounds}$$
$$T_p = \sin\,(\alpha + \beta)\ P_{30} = 0.576 \times 106{,}000 = 61{,}056 \text{ pounds}$$
$$C_{p2} = \frac{c_1}{n}\ C_p = \tfrac{1}{2}\ 86{,}600 = 43{,}300 \text{ pounds.}$$
$$T_{p2} = \frac{c_1}{n}\ T_p = \tfrac{1}{2}\ 61{,}056 = 30{,}528 \text{ pounds.}$$

$$C_w = W \sin\ 30^\circ = \tfrac{1}{2}\ 32{,}000 = 16{,}000 \text{ pounds.}$$
$$T_w = W \cos\ 30^\circ = 0.866 \times 32{,}000 = 27{,}712 \text{ pounds.}$$
$$C_{w2} = \frac{f_2}{m}\ C_w = \tfrac{32}{60}\ 16{,}000 = 8{,}533 \text{ pounds.}$$
$$T_{w2} = \frac{f_2}{m}\ T_w = \tfrac{32}{60}\ 27{,}712 = 14{,}780 \text{ pounds.}$$
$$C_{w3} = \frac{f_1}{m}\ C_w = \tfrac{28}{60}\ 16{,}000 = 7{,}466 \text{ pounds.}$$
$$T_{w3} = \frac{f_1}{m}\ T_w = \tfrac{28}{60}\ 27{,}712 = 12{,}930 \text{ pounds.}$$

$$C_{p2} + C_{w2} = 51{,}833 \text{ pounds.}$$
$$T_{p2} - T_{w2} = 15{,}748 \text{ pounds.}$$

The resultant of the forces C_{w3} and T_{w3} we obtain as

$$V_3 = \frac{f_1}{m}\ W = \tfrac{28}{60}\ 32{,}000 = 14{,}933.$$

The Maximum Strain at the Centre of the Fly-Wheel.

Maximum bending moment

$$M_b = V_3 \times f_2 = 14{,}933 \times 32 = 477{,}856.$$

The section modulus $\dfrac{J}{a} = 0.1\ d^3_4 = 0.1 \times 11^3 = 133.$

Maximum bending strain

$$S_b = \frac{M_b}{J} = \frac{477{,}856}{133} = 3{,}590 \text{ pounds.}$$

The Maximum Strain at the Middle of Journal 2.—
Maximum bending moment

$$M_b = V_3\, m - W\, f_1 = o.$$

Maximum twisting moment

$$M_t = T_p\, r = 61{,}056 \times 16 = 976{,}896.$$

The section modulus for torsion

$$\frac{J_t}{a} = 0.2\, d^3{}_2 = 0.2 \times \overline{9.5}^3 = 171.$$

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{976{,}896}{171} = 5{,}713 \text{ pounds.}$$

Maximum Strain at the Section of the Shaft next to the Right-Hand Crank-Arm.—

The bending moment at bearing 2 being zero, we have, for the section next to the right-hand crank-arm,

Maximum bending moment radially to the crank

$$M_{b1} = (C_{p2} + C_{w2})\left(c_2 - e - \frac{b}{2}\right) = 51{,}833 \times 10 = 518{,}330.$$

Maximum bending moment tangentially to the crank

$$M_{b2} = (T_{p2} - T_{w2})\left(c_2 - e - \frac{b}{2}\right) = 15{,}748 \times 10 = 157{,}480.$$

The combined bending moment becomes

$$M_b = \sqrt{M_{b}^2{}_1 + M_{b}^2{}_2} = \sqrt{\overline{518{,}330}^2 + \overline{157{,}480}^2} = 542{,}000.$$

The section modulus is $\frac{J}{a} = 0.1\, d^3{}_2 = 85.7.$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{542{,}000}{85.7} = 6{,}320 \text{ pounds.}$$

The twisting moment is $M_t = T_p\, r = 976{,}896.$

The section modulus for torsion is $\frac{J_t}{a} = 171.$

Maximum twisting strain

$$S_t = \frac{M_t}{J} = \frac{976,896}{171} \quad 5,712 \text{ pounds.}$$

The combined maximum strain becomes

$$S = 0.35 \times 6,320 + 0.65 \sqrt{\overline{6,320}^2 + 4 \times \overline{5,712}^2} = 10,700 \text{ pounds.}$$

Shaft with Side-Crank.—Let it be assumed that Fig. 70 is a proposed shaft for a 20 × 32 blast-furnace gas-engine.

At the time of the combustion in the cylinder, when the crank

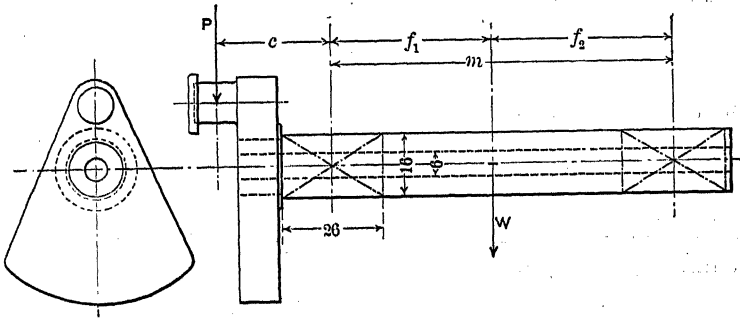



Fig. 70.

passes the centre, the full pressure, P, on the piston will be transmitted to the crank-pin, and the pressure on the main bearing will be $\frac{m+c}{m}P$. The approximate bearing surface of the journal is determined according to equation 110 to suit this pressure and it is found that a projected bearing surface 16″ × 26″ will be ample. In order to obtain a good connection between the crank and shaft it will be required to make the distance c = 28 inches. For a maximum pressure 450 pounds per square inch of the piston, the total pressure on the crank-pin will be P = 141,000 pounds.

According to these data, the strength calculation for the shaft will be as follows:

The Maximum Strain in the Shaft at the Middle Section of the Main Journal.—

Maximum bending moment

$$M_b = P \times c = 141{,}000 \times 28 = 3{,}948{,}000.$$

The section modulus for a hollow shaft

$$\frac{J}{a} = 0.1 \frac{d^4 - d^4_1}{} \quad 0.1 \frac{16^4 - 6^4}{16} = 400.$$

Maximum bending strain

$$S_b = \frac{M_b}{J} = \frac{3{,}948{,}000}{400} = 9{,}870 \text{ pounds.}$$

The section modulus for a solid shaft will be

$$\frac{J}{a} = 0.1\, d^3 = 410.$$

and the maximum bending strain

$$S_b = \frac{3{,}948{,}000}{410} = 9{,}630 \text{ pounds.}$$

When the crank stands 30 degrees past the centre, as Fig. 71, we have $P_{30} = 0.75\, P = 106{,}000$ pounds.

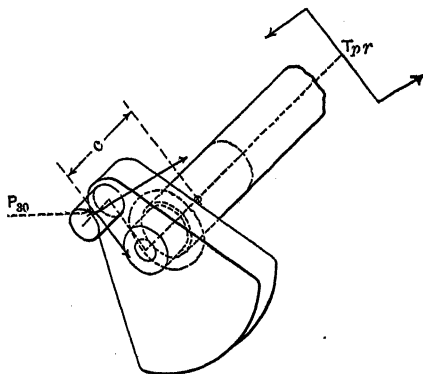



FIG. 71.

The Maximum Strain at the Middle Section of the Journal. Crank 30° Above Centre.—

Maximum bending moment,

$$M_b = P_{30} \times c = 106{,}000 \times 28 = 2{,}968{,}000.$$

Section modulus for a hollow shaft $\frac{J}{a} = 400$.

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{2,968,000}{400} = 7,420 \text{ pounds.}$$

Maximum torsional moment

$$M_t = P_{30}\ r\ sin\ (\alpha + \beta) = 106,000 \times 16 \times 0.576 = 976,900.$$

The section modulus for torsion $\frac{J_t}{a} = 2 \times \frac{J}{a} = 800$.

Maximum twisting strain

$$S_t = \frac{M_t}{\frac{J_t}{a}} = \frac{976,900}{800} = 1,220 \text{ pounds.}$$

The combined maximum strain is

$$S = 0.35 \times 7,420 + 0.65 \sqrt{\overline{7,420}^2 + 4 \times \overline{1,220}^2} = 7,670 \text{ pounds.}$$

Deflection of the Shaft.—When a heavy wheel or generator armature is carried on a long shaft, between the main and outboard journals, it will be necessary to analyze the shaft as to its stiffness.

The deflection of the shaft, expressed in inches, is given by the formula

$$\delta = \tfrac{1}{3} \frac{P}{J E} \frac{f_1^2 \times f_2^2}{m} \text{ inches.} \quad . \quad . \quad . \quad (109)$$

P is the total load carried,
J the moment of inertia of the cross-section $= 0.05\ d^4$,
E the coefficient of elasticity, for steel 30,000,000,
m the distance between the journals, in inches,
f_1 and f_2 the distances from the centre of the wheel, respectively, to the main- and the outboard-bearing.

When the load is applied approximately central between the bearings the equation becomes

$$\delta = \frac{1}{48} \frac{P}{J E} m^3. \quad . \quad . \quad . \quad . \quad . \quad (109a)$$

Assume that a 10-inch shaft, carrying a 32,000-pound wheel,

as shown in Fig. 70, is supported by bearings with 80-inch centres. The weight of the shaft is 2,000 pounds, which added to the weight of the wheel gives the total load 34,000 pounds located centrally between the bearings. The weight of the shaft being small, compared with the weight of the wheel, it may simply be added to the latter weight.

According to the data given, we obtain

$$\delta = \frac{1}{48} \frac{34{,}000 \times 80^3}{0.05 \times 10^4 \times 30{,}000{,}000} = 0.024 \text{ inch.}$$

The deflection of a shaft carrying an electric generator-armature should generally not be allowed to exceed 0.03 inch, and a proper allowance should be made for the magnetic pull on the armature, when below the true centre.

Allowable Strains in the Shaft due to the unreduced maximum pressure on the piston.

The weakest sections of a centre-crank shaft, as generally carried out, are the crank-pin, and the section of the shaft near the crank-arm toward the wheel. In order to avoid the necessity of an excessively heavy connecting-rod, the crank-pin is made small, within safe limits. A maximum strain of 12,000 to 15,000 pounds in the material of the pin can safely be allowed.

The highest strain in the shaft proper should not, however, be more than 11,000 to 13,000 pounds.

There being generally no good reason for making the crank-arms very light, and as the material put into them contributes essentially to the rigidity of the complete shaft, a strain in the arms of 11,000 to 13,000 pounds per square inch is generally not exceeded.

These strains in the shaft may appear heavy, but it will be understood that they do not represent the average maximum working strains but the maximum strains to which the shaft may occasionally be subjected.

CHAPTER XI

ENGINE DETAILS

The Engine-Bed.—Fig. 72 illustrates a type of engine-bed that has been adopted by several builders, for engines of sizes up to 22 inches cylinder-diameter. It is originally of German design and is adapted for a centre-crank shaft.

There are, however, as has been pointed out, some serious objections to the employment of centre-cranks in connection with large twin engines, on account of the necessity of installing and maintaining four or more bearings in alignment for the proper support of the shaft, and on account of the couplings that frequently will be required for coupling the two engine shafts together. The type of engine-bed illustrated in Fig. 73, requiring in any case not more than two journals, is to be preferred for large engines, assuming that there is no serious objection to the expense of a heavy shaft.

The main requirement in an engine-bed is rigidity with respect to the heavy strains that act between the journals and the rear connection to the cylinder. This requirement is, in the engine-beds illustrated, very well provided for, by the employment of deep girder-designs that offer adequate stiffness against the bending forces.

Strain in the Bed.—Fig. 74 represents the journal-end of a side-crank engine-frame. At the crank-pin there is applied a maximum force $P = A \times 450$, which reduced to the centre of the journal becomes P_1.

Let it be desired to find the maximum strain in the section $A\ B$ of an engine-frame for a 24-inch cylinder of the type illustrated, and of the detail-dimensions as given in connection with Fig. 74, on page 255.

The forces acting are:

$P = 452 \times 450 = 203{,}400$ pounds;
$P_1 =$ 220,000 pounds;

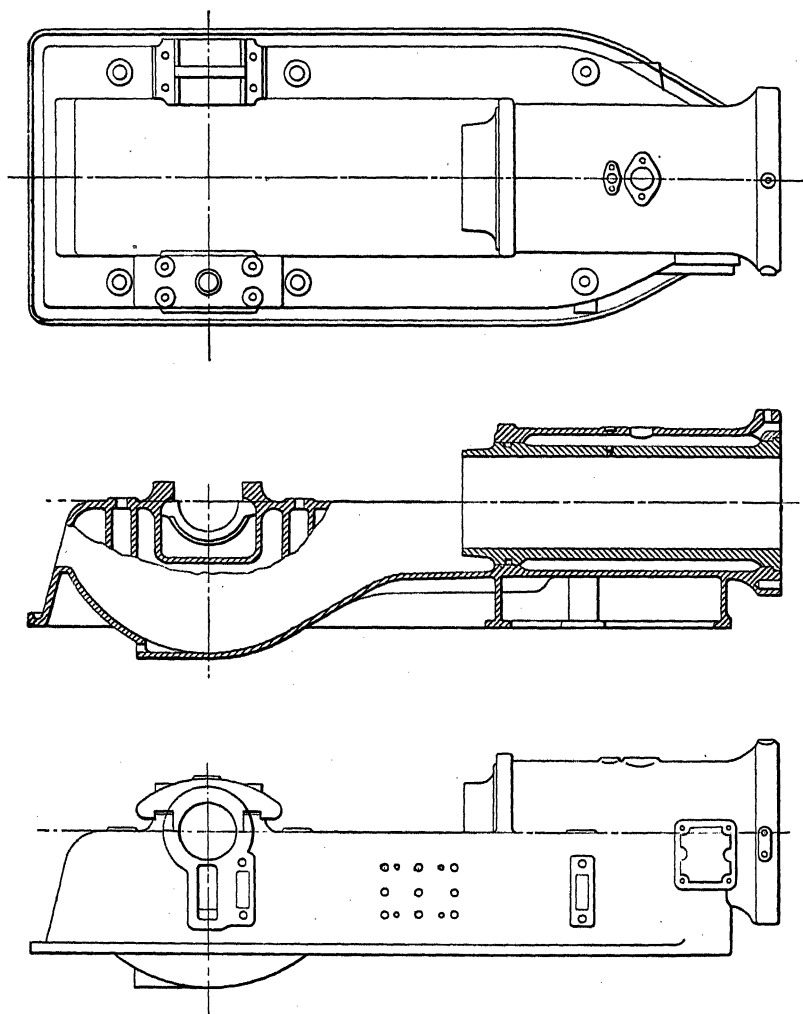



FIG. 72.—Engine-Bed for Centre-Crank Shaft.

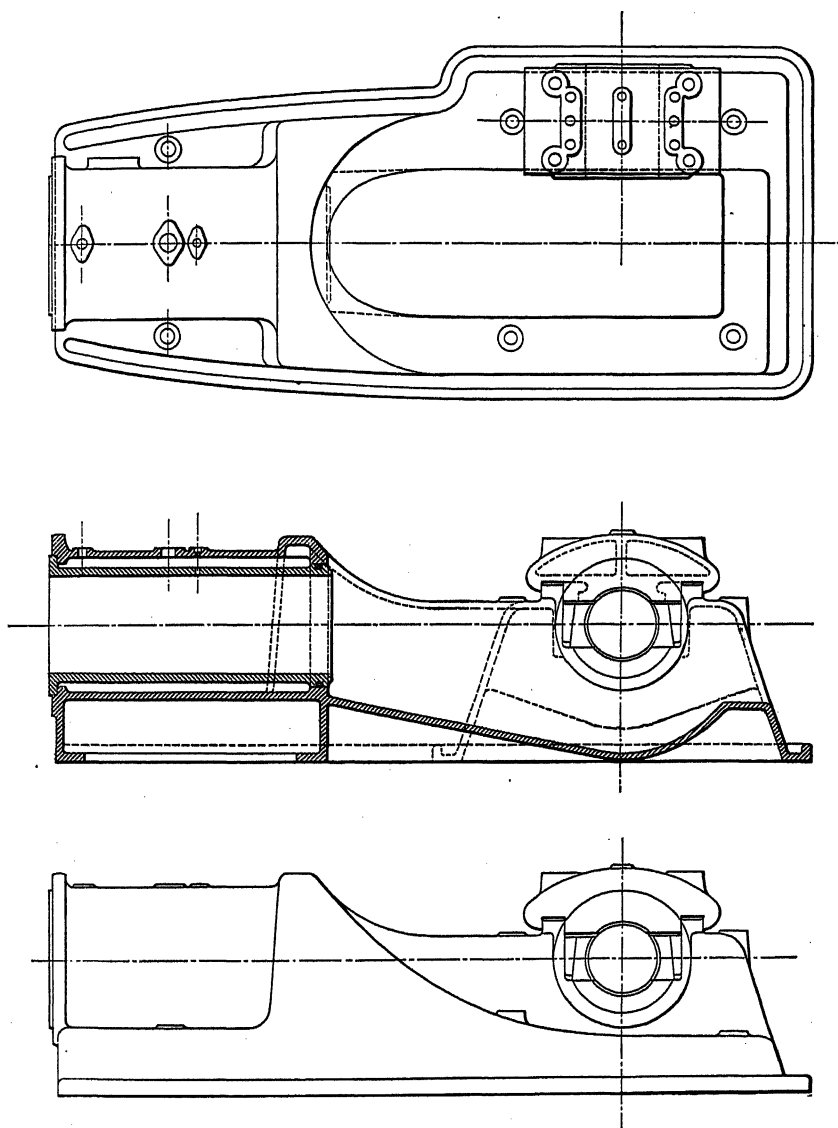



FIG. 73.—Engine-Bed for Side-Crank Shaft.

The bending moment in section $A\ B$ is

$$M_b = P_1 \times l = 220{,}000 \times 11 = 2{,}420{,}000.$$

With reference to the neutral axis $C\ D$:

Moment of inertia of two side walls

$$J_1 = \frac{2\, t_2\, h^3}{12} = \frac{2 \times 1.5 \times 42^{-3}}{12} = 18{,}522.$$

Moment of inertia of top side

$$J_2 = \frac{b\, t_1^3}{12} + t_1\, b\, a_1^2 = \frac{12 \times 1.75^{-3}}{12} + 1.75 \times 12 \times 20.12^{-2} = 8{,}520.$$

Moment of inertia of bottom flanges

$$J_3 = \frac{(c - d - 2t_2)\, t^3_3}{12} + (c - d - 2\, t_2)\, t_3\, a_3^2$$

$$= \frac{(21 - 4 - 3)\, 1.5^{-3}}{12} + (21 - 4 - 3)\, 1.5 \times 20.25^{-2} = 8{,}614.$$

The section modulus of the whole section

$$\frac{J}{a} = \frac{J_1 + J_2 + J_3}{a} = \frac{35{,}656}{21} = 1{,}700.$$

Maximum bending strain

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{2{,}420{,}000}{1{,}700} = 1{,}420 \text{ pounds.}$$

The tensile strain is =

$$\frac{P_1}{\text{area of section}} = \frac{220{,}000}{168} = 1{,}310 \text{ pounds.}$$

Total maximum strain in section $A\ B$,

$$S = 1{,}310 + 1{,}420 = 2{,}730 \text{ pounds.}$$

The Crank-Pin and Piston-Pin Journals.—The maximum pressure per square inch projected surface of the journals, due to the maximum pressure on the piston, can in a gas-engine be allowed higher than what is generally the practice with respect to the steam-engine. This is justified for the reason that the maximum pressure on the gas-engine piston is of short duration

only; hence, the heat evolved by friction is less than in the steam-engine, and an effective lubrication of the journals is more readily maintained. The piston-pin is often made small as a matter of necessity, on account of limited room, but it is with advantage kept as large as circumstances will allow.

The service for which an engine is intended and the grade of the material put into its pins and journal-boxes will, of course, greatly influence their minimum safe dimensions. In practice,

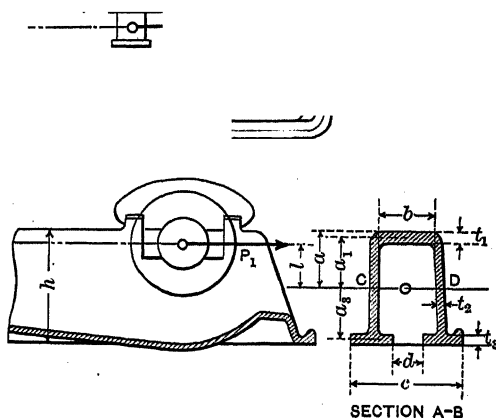



FIG. 74.

therefore, engines required to be as light as possible, and built with particular care, have often pins and bearing-surfaces proportionally much smaller than what would be safe to use in heavier stationary engines. The piston-pin and crank-pin of small engines can, with equal safety, also be made to bear a higher breaking strain, as well as a higher bearing-pressure per square inch projected surface, than engines of medium and high power.

Hence, when considering, in a general way, the limiting size of the wearing surfaces in engines of different construction, distinction should be made, in the first place, between small engines,

and engines of medium and large power. Secondly, between light-built and heavy engines.

The maximum pressures per square inch of projected surface of the journals safely allowed in practice are:

TABLE XXIII.

	On the Main Bearing.		On the Crank Pin.		On the Piston Pin.	
	Heavy Engines.	Light Engines.	Heavy Engines	Light Engines.	Heavy Engines.	Light Engines.
			$B\bar{c}$ Max.		Bw Max.	
Small Engines	400	550	1800	2400	2400	3400
Medium and Large Engines	400	550	1300	1800	1800	2400

The higher figures of the table refer to light-built engines and the lower figures to relatively heavy ones.

The actual maximum pressure on the journals, when the engine is up to speed, is modified very considerably by the inertia of the reciprocating parts, but as a guide for determining the size of the pins the maximum pressure evolved at slow speed is most conveniently used, and this pressure is the basis for the figures in the table above.

It is to be noted, that the maximum pressure on the piston varies in different engines. In engines of high compression of the charge, such as producer-gas or blast-furnace gas engines, the maximum pressure should, with respect to strength and maximum bearing-pressure on the pins, be figured at 450 pounds per square inch, whereas in ordinary, low-compression, gas- or gasoline engines 350 pounds per square inch may be ample. On this account it is necessary to distinguish between the following four classes of engines:

Small engines (below 11 inches cylinder diameter) in which $P_{max.} = 350$ pounds;

Small engines in which $P_{max.} = 450$ pounds;

Medium and large engines in which $P_{max.} = 350$ pounds;

Medium and large engines in which $P_{max.} = 450$ pounds.

A reduction for the inertia of the reciprocating parts can, if desired, readily be made in these pressures, when the weight of the reciprocating parts is known.

To overcome the inertia of the weight G there is required, at the beginning of the stroke, a total pressure on the piston:

$$P_1 = 0.000034\, G\, N^2\, r.$$

With respect to the piston-pin pressure of a trunk-piston engine, G is the weight of the piston and piston-pin; and with respect to the crank-pin pressure, G is the weight of the piston, piston-pin and one-half of the connecting-rod; N is the number of revolutions per minute, and r is the crank-radius, in inches.

The coefficient 0.000034 is, as stated on page 195, only approximate, but for practical purposes it answers very well.

If the force P_1 be subtracted from the total initial pressure on the piston the remainder will be the actual maximum pressure transmitted to the pin at normal speed.

Constructive considerations limit the length of the trunk-piston-pin, l_w, to approximately one-half the cylinder-diameter.

Thus, $$l_w = 0.5\, D.$$

The diameter of the pin, d_w, may also, conveniently, be expressed as a percentage of the cylinder-diameter, as

$$d_w = x_w\, D.$$

Hence, $$l_w\, d_w = 0.5\, x_w\, D^2,$$

and, when $B_{w\ max.}$ is the maximum bearing-pressure allowed per square inch projected surface of the wrist-pin, we have

$$l_w\, d_w\, B_{w\ max.} = \frac{\pi D^2}{4} P_{max.}$$

and $$x_w = \frac{\pi}{2}\, \frac{P_{max.}}{B_{w\ max.}}$$

In a preliminary estimate of the crank-pin for bearing-pressure it will be convenient to assume its length, l_c, to be the same as that of the wrist-pin,

thus $$l_c = 0.5\, D;$$

and if, similarly as before, the pin-diameter, d_c, be expressed in terms of the cylinder-diameter, or

$$x_c D,$$

we get
$$x_c = \frac{\pi}{2} \frac{P_{max.}}{B_{c\ max.}}.$$

The values of x_w and x_c solved from the above equations for the limiting values $B_{w\ max.}$ and $B_{c\ max.}$ of Table XXIII, with respect to light and heavy engines, and for $P_{max.}$ 350 and 450 pounds per square inch, will be found in the following table:

TABLE XXIV.

Limiting Sizes of Pins for Different Classes of Engines.

Piston Pin, $d_w = x_w D$, $l_w = 0.5\ D$.

SMALL ENGINES.

B_w Max.	Light. 3400.		Heavy. 2400.	
P Max. = 350...	$d_w = 0.16\ D$	$S_w = 42700$	$d_w = 0.23\ D$	$S_w = 14000$
P Max. = 450...	$d_w = 0.21\ D$	$S_w = 24400$	$d_w = 0.3\ D$	$S_w = 8330$

LARGE ENGINES.

B_w Max.	Light. 2400.		Heavy. 1800.	
P Max. = 350...	$d_w = 0.23\ D$	$S_w = 14000$	$d_w = 0.3\ D$	$S_w = 6500$
P Max. = 450...	$d_w = 0.3\ D$	$S_w = 8330$	$d_w = 0.39\ D$	$S_w = 3800$

Crank Pin, $d_c = x_c D$, $l_c = 0.5\ D$

	SMALL ENGINES.		LARGE ENGINES.	
B_c Max.	Light. 2400.	Heavy. 1800.	Light. 1800.	Heavy. 1300.
P Max. = 350...	$d_c = 0.23\ D$	$d_c = 0.3\ D$	$d_c = 0.3\ D$	$d_c = 0.42\ D$
P Max. = 450...	$d_c = 0.29\ D$	$d_c = 0.39\ D$	$d_c = 0.39\ D$	$d_c = 0.54\ D$

An approximate estimate of the fibre-stress in the wrist-pin may be obtained by figuring its strength as if it were a circular beam uniformly loaded with the total pressure $\frac{\pi D^2}{4} P_{max.}$, and supported at the ends, immediately outside the journal.

The equation for the safe load on the pin will, accordingly, be:

$$\frac{\pi D^2}{4} P_{max.} = 8 \frac{S_{w\,max.}}{l_w} \frac{J}{a};$$

$S_{w\,max.}$ being the allowable fibre-stress,

$\frac{J}{a}$ the section modulus, which is $= \frac{\pi}{32} d_w^3 = \frac{\pi}{32} x_w^3 D^3$,

and $$l_w = 0.5\,D;$$

whence $$S_{w\,max.} = \frac{1}{2\,x_w^3} P_{max.}$$

By solving $S_{w\,max.}$ for $P_{max.} = 350$ and 450 and for the limiting values of x_w of Table XXIV, the approximate fibre-stress in the corresponding pins may be obtained. These stresses, S_w, are given in the table.

The data regarding the bearing pressure allowed, and the dimensions of the crank-pin, alone, are insufficient for determining the fibre-stress in the pin. This stress cannot, therefore, be given in Table XXIV.

The shearing stress in the two sections of the wrist-pin next to where it is supported in the piston will be obtained by multiplying the strain S_w by the factor x_w.

The shearing stress in the end-sections of a wrist-pin of the dimensions: $d_w = 0.16\ D$, and $l_w = 0.5\ D$, when $P_{max.} = 350$ pounds becomes $0.16 \times 42{,}700 = 6{,}832$ pounds per square inch.

Table XXIV will be of convenience as a guide for determining suitable pin-sizes for an engine. Assuming the pressure on the piston to be, for instance, 350 pounds per square inch, then a wrist-pin of a diameter of 0.16 D, which is the smallest pin used in practice, will carry a bearing-pressure 3,400 pounds per square inch projected surface (neglecting the influence due to acceleration at speed) and the fibre-stress in such a pin would be approximately 42,700 pounds. Under the same conditions, a wrist-pin of a diameter 0.3 D – a large pin, will carry a bearing-pressure of 1,800 pounds per square inch, and the approximate fibre-stress will be 6,500 pounds. Similarly, a crank-pin of a diameter 0.23 D will carry a bearing-pressure of 2,400 pounds, and a crank-pin of a diameter 0.42 D a pressure of 1,300 pounds per square inch projected surface.

The intensity of the bearing-pressure on the wrist-pin being occasionally very high it is necessary that the pin be made of a hard material, or, when made of mild steel, it must be case-hardened.

A piston-pin of a diameter of 0.16 of the cylinder-diameter and of a length of 0.5 the cylinder-diameter, carrying a working stress of 42,700 pounds, figured due to the bending of the pin, and a stress 6,832 pounds, figured for shear at the ends, would of course be considered altogether too light. Such pin-diameters are, however, sometimes found in practice, but the length must be made smaller than ½ D; the bearing-pressure, thus, allowed higher than 3,400 pounds as assumed in the table. The length of the pin is rarely actually made over 1¾ to 2 times its diameter.

Main Shaft Journals.—In determining the size of the main journals of a gas-engine, it is convenient to consider the load on the bearings as resulting in two different ways:

First, the load due to the maximum pressure on the piston may be assumed to be transported directly to the bearing, without any reduction for inertia forces, in which case the maximum intensity of the pressure per square inch projected surface becomes $\frac{P_{max.} A}{l\,d}$. As this force acts only during a short time, under the best conditions for free lubrication, and as it actually, under normal speed, will be considerably reduced due to inertia forces, it can be allowed as high as between 275 to 550 pounds per square inch.

Thus, generally, $\frac{P_{max.} A}{l\,d} < 550$ pounds. . . . (110)

Secondly, the total pressure on the journal may be considered as made up, partly, of the forces transmitted to the shaft from the variable pressure on the piston during the entire cycle, and, partly, of the constant vertical pressure which is due to the weight of the shaft, fly-wheel or generator-armature.

The mean intensity, P_h, of the forces acting on one side or the other of the journal, due to the pressure on the piston, will, of course, vary with the number, and arrangement, of the cylinders. It will, on an average, be

$$P_h = \frac{F\,A}{l\,d} \quad . \quad . \quad . \quad . \quad . \quad . \quad (111)$$

The coefficient F having the following values:

in a four-cycle, single-acting single-cylinder engine $F = 40$
in a four-cycle, single-acting two-cylinder opposed engine $F = 60$
in a four-cycle, single-acting two-cylinder tandem engine $F = 70$
in a four-cycle, double-acting one-cylinder engine $F = 60$
in a four-cycle, double-acting two-cylinder tandem engine $F = 100$
in a two-cycle single-cylinder engine $F = 110$

A is the area of one cylinder of one engine, in square inches; l is the length of one bearing, and d its diameter, in inches. A twin engine is only a duplication of a single engine, and the average pressure in the bearing will be the same as in the single engine of the same type.

For a two-cylinder engine with three bearings, or a three-cylinder engine with four bearings, l will be the length of the journal which takes the greatest load due to the weight of the wheels, etc., generally the outside journal.

If V is the pressure on the journal due to the weight of shaft, wheels, etc., the mean pressure per square inch of projected surface of the journal becomes $P_v = \frac{V}{l\,d}$ . . . . (112)

The heating of the journal is caused by the pressures P_h and P_v, and it will depend also on the surface-speed between the shaft and the bearing.

For cool running, with ordinary lubrication, it is required that

$$P_h + P_v = \frac{\frac{1}{2} F A + V}{d\,l} < \frac{700}{\sqrt{d\frac{N}{60}}}; \quad . \quad . \quad (113)$$

d being the diameter of the journal in inches, and

$\frac{N}{60}$ the number of turns per second.

Main Bearings.—The main bearings of engines up to twenty inches cylinder-diameter are generally provided with babbitt-lined, removable shells, divided on the horizontal line. The construction of an ordinary bearing of this type is shown in Fig. 75. For larger engines adjustable quarter-boxes, as shown in Fig. 73, would be preferable. One quarter-box may be sufficient, par-

ticularly in single-acting engines, and it is properly placed on the side toward the cylinder, opposite the side on which the principal pressure acts.

For the outboard bearing, which carries only a vertical load,

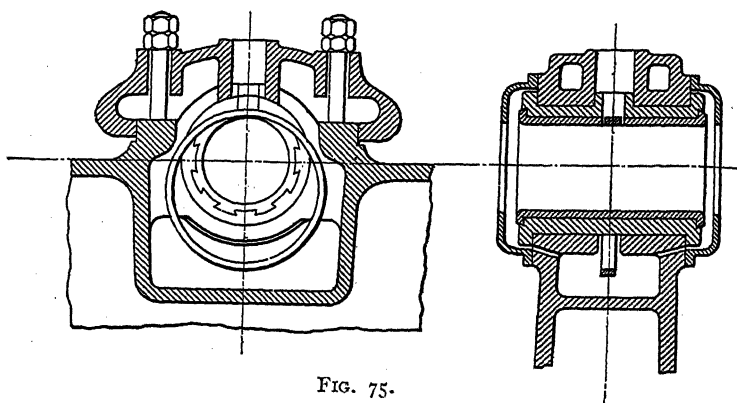



Fig. 75.

no loose boxes are, as a rule, provided. This bearing is, however, often placed on a sole-plate provided with adjusting screws to facilitate the horizontal adjustment of the bearing, at erection, or whenever adjustment would become necessary.

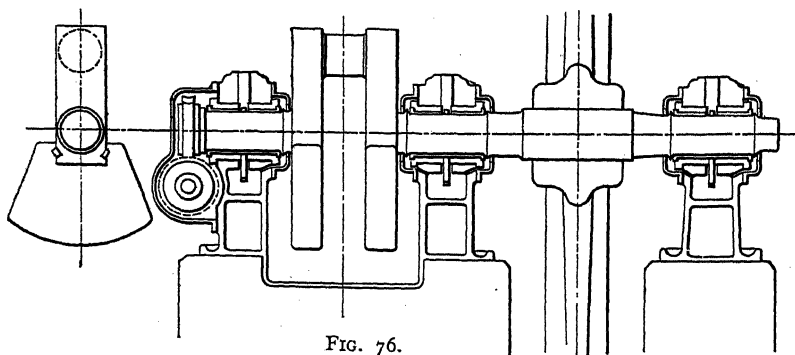



Fig. 76.

Lubrication.—The ring-oiling system of lubrication is simple and effective, but great care should be taken to design the ring and channel for the same so that the ring will run perfectly free, and so that there will be no liability for it to stick. Account

should be taken of the fact that the ring swings out of the plumb-line, when revolving.

For long journals two rings are generally used, placed on a distance of one-third of the length of the journal apart. The rings are made in halves, hinged together, and the open joint dovetailed together, and secured in some way so that it cannot, on any account, open up and get stalled. Large rings should be made of some hard material, that is not readily bent out of shape when the ring is put in place.

Fig. 76 shows an arrangement for the oiling of the main journals, in detail.

The principal trouble experienced with ring-oiling bearings

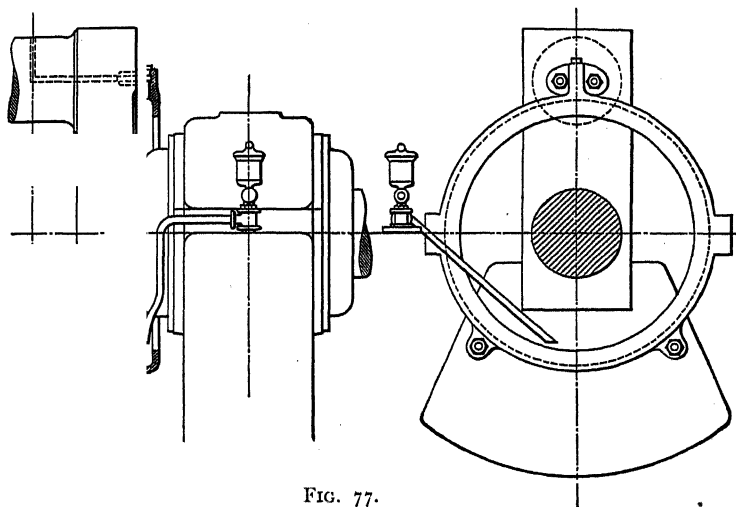



FIG. 77.

has been due to the unexpected stalling of the rings, from one cause or other. In a large plant, a continuous automatic oiling system is, of course, the most modern and most reliable.

The oiling of the crank-pin of a centre-crank is often effected as shown in Fig. 77. The arrangement consists of a cup-ring surrounding the shaft, which catches the oil delivered by the sight-feed cup, and carries it by the action of the centrifugal force through the oil pipe to the crank-pin.

A common oiling arrangement for the piston-pin is illustrated

in Fig. 78. The oil is caught by the extended wiper-cup and carried back to the oil-receptacle formed in the top of the rod-end.

The Piston.—The highest normal pressure between the piston and the cylinder, due to the reaction from the connecting-rod, occurs when the crank stands at an elevation of approximately

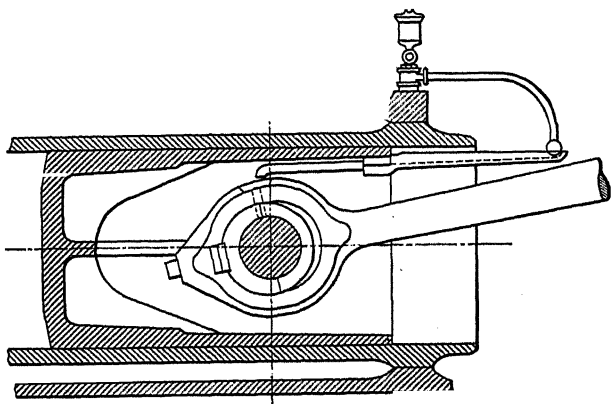



FIG. 78.

30 degrees from the head-end centre. This pressure is, in Fig. 79, designated N.

For the ratio $\frac{r}{l} = \frac{1}{5.5}$, and $\alpha = 30°$, we have

$$\beta = 5° - 10' \quad \text{and} \quad \tan \beta = 0.09.$$

As shown, page 228,

$$P_{30} = 0.75\, P, \text{ approximately,}$$

thus $N_{max.} = 0.09\, P_{30} = 0.09 \times 0.75\; P = 0.068\, P.$

The effective bearing surface of the piston may be counted as being

$$F = 0.85\, D \times 0.8\; L = 0.68\, D\, L.$$

Hence, the bearing pressure per square inch of effective piston-surface

$$p_n = \frac{0.068\; P}{0.68\; D\, L} = \frac{P}{10\, D\, L}.$$

This pressure, exclusively of the weight of the piston, should not exceed 25 pounds,

thus $$p_n = 0.1 \frac{P}{DL} < 25$$

and $$D \times L > \frac{1}{250} P.$$

If $P = 450 \frac{\pi D^2}{4}$ it will be required that $\frac{L}{D} > 1.41$;

and if $P = 350 \frac{\pi D^2}{4}$ it will be required that $\frac{L}{D} > 1.09$. (114)

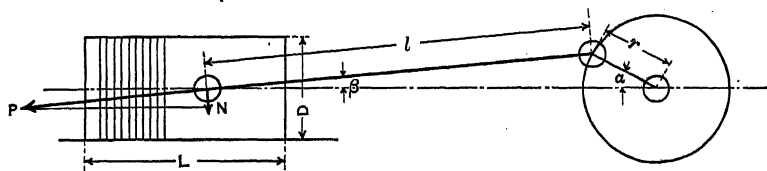



FIG. 79.

In a 20 × 30 engine with high compression we have

$$P = 314 \times 450 = 141,300 \text{ pounds},$$

and $$N_{max.} = 0.068 \times 141,300 = 9,608 \text{ pounds},$$

$D = 20$,

$L = 30$,

thus $$F = 0.68\ DL = 408 \text{ square inches.}$$

Hence $$p_n = \frac{9,608}{408} = 23.5 \text{ pounds.}$$

The Strength of the Piston.—In order to give stiffness to the piston it should be heavily ribbed, from the middle of the bottom some distance down its side, as in Fig. 80. The bending strain in the flat bottom, between the ribs and the circular wall, of a radius approximately ¼ D will be

$$\tfrac{1}{16}\, p \frac{D^2}{t^2}, \qquad (115)$$

when p is the pressure per square inch of the piston, D its diameter in inches, and t the thickness of its bottom.

The proper bending strain to allow in this case, when the pressure per square inch of the piston is figured $p = 450$ pounds, would be $S = 3,500$ pounds.

Thus, $$3{,}500 = \frac{1}{16}\, 450\, \frac{D^2}{t^2},$$

or $$t = 0.09\, D.$$

Hence, the piston for a 20 × 30 cylinder requires a thickness $t = 0.09 \times 20 = 1\frac{7}{8}$ inch; the material being strained 3,500 pounds.

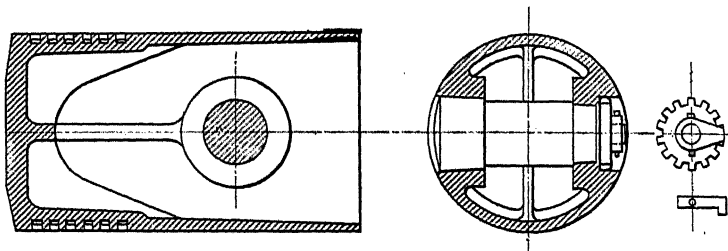



FIG. 80.

When the bottom is made curved to a radius $r = 2\, D$ the thickness of the bottom may be made $t = 0.075\, D$.

The Strength of the Piston Pin.—The pins for large pistons are most conservatively figured, for strength, as if supported at the mid-sections of their bearings in the trunnion bosses. Considering the load from the connecting-rod to be evenly distributed over the pin, we get, according to Fig. 81, the bending moment.

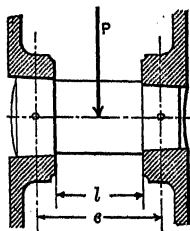



FIG. 81.

$$M_1 = \frac{P}{2}\left(\frac{c}{2} - \frac{l}{4}\right).$$

The section modulus is 0.1 hence the maximum strain at the middle of the pin

$$S_b = 2.5\, \frac{P}{d^3}\left(c - \frac{l}{2}\right). \quad . \qquad (116)$$

For a 20-inch engine:

$P = 141{,}300$ pounds,
$d = 7\frac{1}{2}''$,
$l = 9\frac{1}{2}''$,
$c = 13\frac{3}{8}''$.

Thus $$S_b = 2.5 \frac{141,300}{422}(13.375 - 4.75);$$

$$S_b = 7,200 \text{ pounds.}$$

The Piston Ring.—The open-end piston is generally provided with a number of spring-rings in order to reduce the pressure with which each ring must be sprung against the cylinder; 5 to 7 rings are often employed, exerting, each, a pressure against the cylinder of 4½ to 3½ pounds per square inch.

The thickness of the ring and the amount which has been cut out of the complete ring is of course what determines its stiffness when in place.

When a cast-iron ring, D inches in diameter, is sprung together a distance e the maximum strain in the middle section will be

$$S_b = \frac{e\ t\ E}{2.4\ D^2}; \quad . \quad . \quad . \quad . \quad . \quad . \quad (117)$$

t being the thickness of the ring, and E the modulus of elasticity, for cast-iron = 12,000,000. And the tension per square inch of ring-surface will be

$$q = \frac{S_b}{3}\left(\frac{t^2}{D^2}\right),$$

or $$S_b = 3\ q\ \frac{D^2}{t^2}.$$

If we allow a strain S_b = 11,000 pounds, and require the spring-tension to be q = 3½ pounds per square inch, we obtain

$$t = \frac{D}{32}.$$

In springing the ring over the piston there will be exerted a bending strain in the fibres

$$S_b = \frac{E\ t^2}{2.5 \times 0.25\ D^2} = 19,200,000\ \frac{t^2}{D^2},$$

which for $t = \frac{D}{32}$ becomes

$$S_b = 19,000 \text{ pounds.}$$

The width of the ring is made, for small engines, ⅜ to ½ inch, and ⅝ to ¾ for large ones.

Fig. 80 shows the general construction of an open-end piston for a cylinder of medium size. The piston-pin nut is slotted for a spanner, and it is locked by means of a keeper, which is fitted to a spanner notch, and secured to the wrist-pin by means of a taper pin driven through the eye of the keeper and through a prolongation of the wrist-pin proper.

The piston should be fitted $\frac{1}{1000}$ inch free in the cylinder, per inch diameter, for its general length, and its extreme inside end is generally turned off $\frac{1}{64}$ to $\frac{1}{32}$ inch free, tapering to about 1½ to 3 inches forward. The smaller figures apply to the case of smaller pistons and the larger ones to medium sizes, of 12 to 20 inches in diameter.

The Connecting-Rod.—The main body of the connecting-rod is, for strength, treated as a hinged strut subjected to a compressive force equal to the total pressure on the piston. The load it can safely carry is computed according to the formula

$$fP = \frac{\pi^2 E J}{l^2}, \qquad (118)$$

in which

P is the safe compressive load;

f the factor of safety;

E the modulus of elasticity = 30,000,000 for steel;

J the moment of inertia of the middle section of the rod

$$= 0.05\ d^4 \text{ for a circular rod}$$

$$= \frac{b\ h^3}{12} \text{ for a rectangular section;}$$

d the diameter of the middle section, in inches;

l the length of the rod, in inches.

For a 20 × 32 engine

$$P = 141,300,$$

$$J = 0.05 \times 4.75^{-4} = 25.5,$$

$$l = 80.$$

Hence,

$$f = \frac{9.86 \times 30,000,000 \times 25.5}{141,300 \times 6,400};$$

$$f = 7.5.$$

Wrist-Pin End of Rod.—The force required for the acceleration

of the piston and piston-pin at the head-end centre causes the maximum tensile strain in the eye of the wrist-pin end of the rod.

If the weight to be accelerated is G_2, we have, according to equation 101,

FIG. 82.

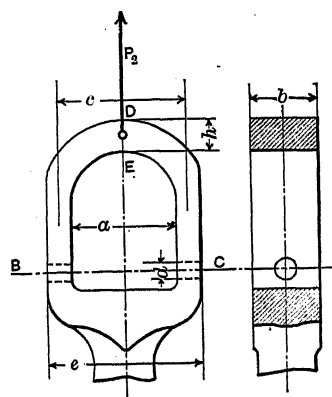
$$P_2 = 0.000034\, G_2\, N^2\, r.$$

P_2 being the accelerating force at the head-end centre,

N the number of revolutions per minute, and

r the crank-radius in inches.

The area of the cross-section through the eye of the rod must be made to resist, in tension, the force P_2.

It will not, generally, be necessary to strain the material higher than at 5,000 pounds.

The bending strain on the back of the eye, at section $D\,E$, Fig. 82, will be:

Maximum bending moment $M_b = \dfrac{P_2\, c}{4}$,

Section modulus $\dfrac{J}{a} = \dfrac{b\, h^2}{6}$,

Maximum bending strain $S_b = \dfrac{1.5\, P_2}{b\, h^2}$

In a 20 × 32 engine we have:

Weight of piston	730 pounds.
Weight of piston-pin	180 pounds.
Total	910 pounds.

Hence $P_2 = 0.000034 \times 910 \times 25{,}600 \times 16$, at 160 revolutions.

$P_2 = 12{,}670$ pounds.

Other data are:

$a = 9\frac{1}{4}''$, $b = 6\frac{1}{4}''$, $c = 11''$, $d = 1\frac{3}{8}''$, $e = 13\frac{1}{4}''$, $h = 2\frac{3}{4}''$.

Thus the area of section $B\,C$

$$F = (e - a)\,(b - d)$$
$$= (13.25 - 9.25)\,(6.25 - 1.375) = 19.5 \text{ square inches.}$$

The tensile strain in this section, therefore,

$$S = \frac{P_2}{F} = \frac{12,670}{19.5} = 650 \text{ pounds.}$$

The bending strain at $D E$ is

$$S_b = \frac{1.5 \times 12,670 \times 11}{6.25 \times 7.56} = 4,420 \text{ pounds.}$$

Crank-End of Rod.—The maximum strain on the crank-end cap and cap-bolts is due to the force required for the acceleration of the reciprocating parts when the crank passes the head-end centre.

FIG. 83.

If G is the weight of the reciprocating parts (the piston and piston-pin and one-half of the connecting-rod), the accelerating force at the head end centre is

$$P_1 = 0.000034\, G\, N^2\, r;$$

N being the number of revolutions per minute and
r the crank radius in inches.

If the area at the bottom of the thread of two cap-bolts is $2a$,

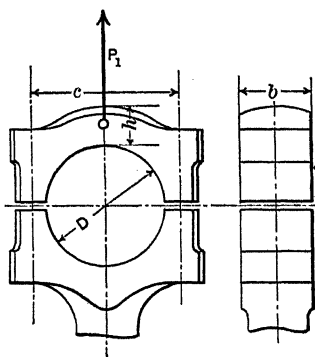
then the maximum tensile strain is $\frac{P_1}{2\,a}$ per square inch.

The bending strain in the rod-end cap:

If the dimensions of the cap are as shown in Fig. 83, the bending moment due to P_1, evenly distributed, becomes

$$M_h = \frac{P_1}{4}\left(c - \frac{D}{2}\right).$$

The section modulus for the middle section of the cap is

$$\frac{J}{a} = \frac{b\,h^2}{6}.$$

The maximum bending strain in cap

$$S_b = \frac{M_b}{\frac{J}{a}} = \frac{1.5\ P_1}{b\ h^2}\left(c - \frac{D}{2}\right). \quad . \qquad (119)$$

For a 20 × 32 engine at 160 revolutions:

The weight of 20″ piston	730 pounds.
The weight of piston-pin	180 pounds.
½ the weight of connecting-rod	530 pounds.
Total	1440 pounds.

Hence $P_1 = 0.000034 \times 1{,}440 \times 25{,}600 \times 16 = 20{,}054$ pounds.

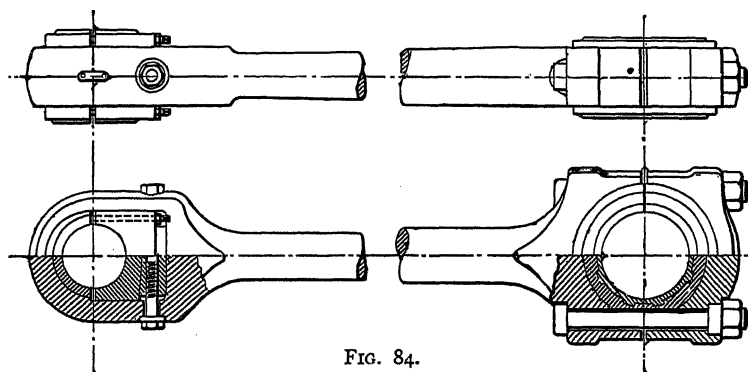



Fig. 84.

If two 2-inch cap-bolts be used (the area at bottom of thread 2 × 2.3 square inches), the maximum strain in the bolts due to inertia becomes

$$\frac{20{,}054}{4.6} = 4{,}360 \text{ pounds per square inch.}$$

Bending strain in the cap:

The distance between bolts,	$c = 12.75''$.
Bore of cap	$D = 12.5''$.
Thickness of cap	$h = 3.0''$.
Width of cap	$b = 6.0''$.

Maximum bending strain

$$S_b = \frac{1.5 \times 20{,}054}{6 \times 9}(12.75 - 6.25) = 3{,}620 \text{ pounds.}$$

Should the shell bear on the cap at the middle only, the strain becomes $2 \times 3{,}620 = 7{,}240$ pounds.

A commonly employed design for a connecting-rod for medium-sized engines is shown in Fig. 84, and an alternate design of the wrist-pin end in Fig. 85.

The Strength of the Fly-Wheel.—Disregarding the influence of the arms on the strength of a fly-wheel ring, the tensile stresses

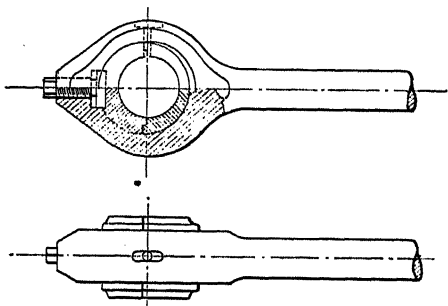
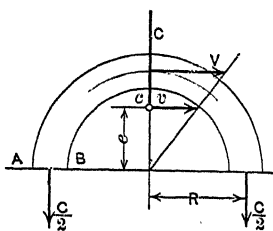



FIG. 85. FIG. 86.

which act in each section, A B, of the ring, Fig. 86, are ½ C, when C is the centrifugal force due to one-half of the weight of the ring.

The centrifugal force, applied at the centre of gravity c, is

$$C = \frac{W}{2g} \frac{v^2}{e},$$

when W is the weight of the complete rim, in pounds, and v the velocity of the point c, in feet per second.

But as $$e = \frac{2R}{\pi}, \quad \text{and} \quad v = \frac{2\pi e N}{60},$$

we get $$C = \frac{W}{g} \frac{\pi R N^2}{900} = 0.00010848 \; W R N^2,$$

and the stress in each section of the rim.

$$\tfrac{1}{2} C = \frac{W}{2g} \frac{V^2}{\pi R} = 0.00005424 \; W R N^2; \quad . \quad (120)$$

V being the velocity of the centre of the wheel-rim, in feet per second.

Assuming the rim-velocity to be 100 feet per second = 6,000

feet per minute, which is the highest velocity generally allowed in cast-iron wheels;

A the area of the rim-section in square feet ($= 144\,A$ square inches), then $W = 2\,A\,\pi\,R\,450$ pounds for a cast-iron rim, and $\frac{1}{2}\,C = \frac{4,500,000\,A}{g}$.

Thus, the strain per square inch of the material is practically 1,000 pounds.

When a wheel is made in halves, the links joining the halves together should not be strained higher than 7,000 to 8,000 pounds per square inch, when they are submitted to direct tension only. The rule is, therefore, to make the smallest cross-section of the links for one joint 12½ to 14½ per cent of the section of the wheel-rim. When the links, or bolts, making the joints are not located centrally to the centre of gravity of the section, the bending moment acting in the joint must be considered, and the area of the bolts and links increased accordingly.

Fig. 87 shows the design for a wheel made in halves in which ordinary T-head links are used for shrinking the rim-halves together. In the hub are used bolts, and these should be made of a liberal diameter, as, in shrinking them in place, they are put to a strain which may be considerable. The diameters of the hub-bolts given in Table XXVIII have been found satisfactory in practice.

One key only should be used in the hub, as it is much easier to fit one large key well than to fit two smaller ones.

Tables of Practical Data Pertaining to Four-Cycle Single-Acting Gas-Engines.—The data contained in the following Tables, XXV and XXVI, referring to the power and dimensions of a line of medium-size producer-gas engines are revised from a line of engines in successful operation. The power and principal data given may be considered quite conservative.

Table XXV contains data with reference to the power, speed, and expected efficiency.

Table XXVI contains the principal dimensions of centre-crank shafts, the general design of which is shown in Fig. 88.

Tables XXVII and XXVIII give the weight and principal

TABLE XXV.

Dimensions and Power of Producer-Gas Engines.

Piston Diameter	12	13	14	15	16	17	18	19	20	21	22
Stroke	18	20	20	24	24	28	28	28	32	32	32
Revolutions per minute	220	210	210	200	200	180	180	180	160	160	160
Piston speed, feet per minute	660	700	700	800	800	840	840	840	853	853	853
Piston area, square inches	113.1	132.7	153.9	176.7	201.1	227.0	254.5	283.5	314.2	346.4	380.1
M.E.P.	69	70	70	70	70	71	71	71	72	72	72
Maximum I.H.P.	39	49	57	75	85	102	115	128	146	161	177
Maximum B.H.P.	32	40	47	63	71	86	98	108	124	137	150
Rated B.H.P.	28	35	41	55	62	75	85	95	110	120	130
Mechanical efficiency	83	83	83	84	84	85	85	85	85	85	85

Maximum I. H. P. $= \frac{P_{mc} L A N}{2 \times 33,000}$. Maximum B. H. P. $= m \frac{P_{mc} L A N}{2 \times 33,000}$.

TABLE XXVI.

Dimensions of Centre-Crank Shafts.

Size of Engine	d	l	d_1	l_1	d_3	l_3	d_4	b	h	c
12×18	5½	5½	5	9½	5	10	6½	3½	7	12½
13×20	6¼	6¼	5½	11	5½	11	7	4	7½	13¾
14×20	6¼	6¼	5½	11	5½	11	7	4½	8	14¾
15×24	7	7	6½	13	6½	13	8	4½	8	15¾
16×24	7	7	6½	13	6½	13	8	4½	8	15¾
17×28	8½	8½	8	15	8	16	9½	5½	9	18¾
18×28	8½	8½	8	15	8	16	9½	5½	9	18¾
19×28	8½	8½	8	15	8	16	9½	5½	9	18¾
20×32	10	10	9½	18	9½	19	11	6½	10½	21½
21×32	10	10	9½	18	9½	19	11	6½	10½	21½
22×32	10	10	9½	18	9½	19	11	6½	10½	21½

The letters at head of columns refer to dimensions as denoted in Fig. 88.

TABLE XXVII.

Diameter, Speed, and Weight of Fly-Wheels.

Size of Engine.	Rated Brake Horse Power.	Number of Revolutions.	Diam.	Rim-Speed.	Weight of "Standard Wheel" $K = 35$.			Weight of "Extra Heavy Wheel" $K = 70$.		
					Rim. Lbs.	Hub and Arms. Lbs.	Total. Lbs.	Rim. Lbs.	Hub and Arms. Lbs	Total. Lbs.
12×18	28	220	7′–6	5183	2400	1500	3900	4800	1600	6400
13×20	35	210	8′–0	5277	3100	1600	4700	6200	2000	8200
14×20	41	210	8′–0	5277	3600	1600	5200	7200	2000	9200
15×24	55	200	9′–0	5654	4700	2600	7300	9400	3000	12400
16×24	62	200	9′–0	5654	5300	2600	7900	10600	3000	13600
17×28	75	180	10′–0	5657	6400	3500	9900	12800	4400	17200
18×28	85	180	10′–0	5657	7200	3500	10700	14400	4400	18800
19×28	95	180	10′–0	5657	8100	3500	11600	16200	4400	20600
20×32	110	160	11′–0	5530	10100	5200	15300	20200	6400	26600
21×32	120	160	11′–0	5530	11100	5200	16300	22200	6400	28600
22×32	130	160	11′–0	5530	12200	5200	17400	24400	6400	30800

TABLE XXVIII.

Principal Dimensions for the Wheels of Table XXVII.

Diameter.	Standard Wheel.						Extra Heavy Wheel.						Hub.			
	Weight.	*h*	*w*	*a*	*b*	No. of Arms.	Weight.	*h*	*w*	*a*	*b*	No. of Arms.	*c*	*d*	*l*	Bolt Diam.
7′-6	3,900	6	6	7½	3¾	6	6,400	8½	8½	8	4	6	6½	14	13	1¾
8′-0	4,700	6¼	7	8	4	6	8,200	9	9¾	9½	4¾	6	7	15	14	1¾
8′-0	5,200	6¼	8	8	4	6	9,200	9	11½	9½	4¾	6	7	15	14	1¾
9′-0	7,300	7½	8	9	4½	6	12,400	10	12¼	10	5	6	8	18	16	2
9′-0	7,900	7½	8¾	9	4½	6	13,600	10	13	10	5	6	8	18	16	2
10′-0	9,900	8½	8½	10	5	6	17,200	11	13¼	12	6	6	9½	21	19	2½
10′-0	10,700	8½	9½	10	5	6	18,800	11	15	12	6	6	9½	21	19	2½
10′-0	11,600	8½	10½	10	5	6	20,600	11	16½	12	6	6	9½	21	19	2½
11′-0	15,300	9½	10½	11½	5¾	6	26,600	12	16½	13½	6¾	6	11	24	22	2¾
11′-0	16,300	9½	11½	11½	5¾	6	28,600	12	18¾	13½	6¾	6	11	24	22	2¾
11′-0	17,400	9½	12½	11½	5¾	6	30,800	12	20	13½	6¾	6	11	24	22	2¾

A suitable taper of the arms is:
For the wide side $\frac{3}{8}$ inch per foot, over all.
For the narrow side $\frac{3}{16}$ inch per foot, over all.

The letters at head of columns refer to dimensions as denoted in Fig. 87.

RIM-JOINT LOOKING FROM x

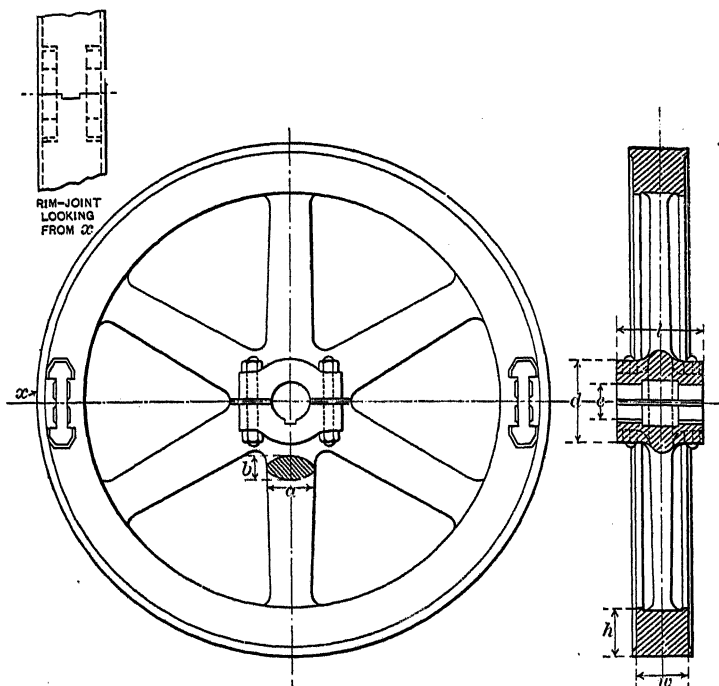



FIG. 87.—Fly-Wheel. For principal dimensions for various classes of engines see Tables XXVII and XXVIII.

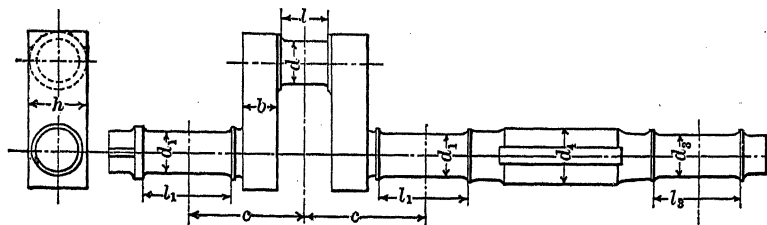



FIG. 88.—Centre-Crank Shaft. For dimensions suitable for various sizes of engines see Table XXVI.

dimensions of two types of fly-wheels generally used in gas-engine practice. They are designated by the names "Standard Wheels" and "Extra Heavy Wheels." The former are suitable for general mill service (the coefficient of steadiness $K = 35$); the latter is required for electric light and power service ($K = 70$).

The sizes of crank-pins and piston-pins, Table XXIX, have been determined by the methods explained in the preceding, and they conform closely with good general practice.

The bearing-pressures on the pins contained in the 6th and 10th columns of the table are the maximum pressures that occur generally only under very good conditions, corresponding to 450 pounds per square inch of the piston. The pressure on the piston under ordinary conditions is, on an average, not over 350 to 400 pounds, and, hence the bearing-pressures on the pins corresponding to these figures are $\frac{7}{9}$ to $\frac{8}{9}$ of those given in the table.

As the strength of all pins must be figured to suit the maximum pressure that is likely to occur, and the bearing-pressure on the pins being generally determined in connection with their strength, both the strength and bearing-pressure per square inch projected surface have been referred to the maximum pressure 450 pounds per square inch of the piston.

Automatic Valves.—Only small unimportant engines have inlet-valves automatically operated by the suction of the piston, as the simple means whereby the valves can be operated mechanically amply justifies its adoption. It is evident that automatically operated valves must be particularly unsuitable for engines running at a high speed, because the higher the speed, and the quicker the valve must open and close, the heavier springs it is necessary to apply back of the valves, to overcome the inertia in moving them. The pressure difference between the inside of the cylinder and the outside supply must, therefore, in high-speed engines, be considerable before the valves lift. This causes loss, both in efficiency and in capacity of the engine. The former since the piston must move against a considerable suction-resistance, the latter due to the low density of the charge in the cylinder at the completed suction-stroke. Engines built with the object in view of obtaining the highest possible power in a

TABLE XXIX.

Size of and Bearing Pressure on Crank-Pins and Piston-Pins.

Size of Engine.	Total Pressure on Piston at 450 lbs. per sq. in.	CRANK-PIN.				PISTON-PIN.			
		Diameter.	Length.	Projected Area.	Pressure per sq. in.	Diameter.	Length.	Projected Area.	Pressure per sq. in.
12 x 18	50,800	5½	5½	30.25	1,680	4½	5½	24.75	2,050
13 x 20	59,800	6¼	6¼	39.00	1,530	4¾	6½	30.87	1,930
14 x 20	69,300	6¼	6¼	39.00	1,770	4¾	6½	30.87	2,240
15 x 24	79,600	7	7	49.00	1,630	5½	7½	41.25	1,930
16 x 24	90,400	7	7	49.00	1,840	5½	7½	41.25	2,190
17 x 28	102,000	8½	8½	72.25	1,410	6¼	8½	53.12	1,920
18 x 28	114,000	8½	8½	72.25	1,580	6¼	8½	53.12	2,140
19 x 28	128,000	8½	8½	72.25	1,770	6¼	9	56.25	2,280
20 x 32	141,000	10	10	100.00	1,410	7½	9½	71.25	1.980
21 x 32	155,000	10	10	100.00	1,550	7½	10	75.00	2,060
22 x 32	171,000	10	10	100.00	1,710	7½	10½	78.75	2,170

cylinder of small diameter, even if the efficiency should be of minor importance, are, therefore, always provided with mechanically moved valves.

To ascertain that the valve-setting is correct and that the valve-ports are adequate for the speed at which an engine is operating, weak-spring cards, that show the vacuum created on the suction-stroke and the back pressure during the exhaust, are generally taken in connection with engine tests. Fig. 89 is such a card taken with a 20-pound spring. The compression line starts from the end of the suction line *a a* at an initial pressure $1\frac{3}{8}$ pounds below the atmospheric line *A B* and it discontinues at *b*, which is as far as the instrument allows the spring to be compressed. The expansion line is not indicated in the card, excepting the very lowest end of it; the release taking place soon after the line becomes visible at *c*. The exhaust-line *d d* is approximately 2 pounds above the atmospheric line.

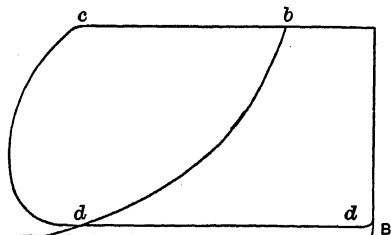



FIG. 89.

Inlet- and Exhaust-Valves.—The variation in the velocity

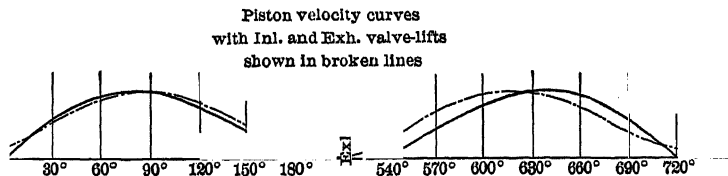



FIG. 90.

of the piston during the forward and return stroke is shown by the curve, Fig. 90. This curve has been obtained by plotting at the successive crank-positions the length of the ordinates given by the equation for the velocity of the piston-pin (see page 471).

$$S = V\left(\sin a + \tfrac{1}{2}\,\frac{r}{l}\,\sin 2\,a\right),$$

in which

S is the piston-velocity, in feet per second;

V the linear velocity of the crank-pin, in feet per second $\frac{2\pi r n}{60}$;

α the crank-angle, counted from the head-end centre;

$\frac{r}{l}$ the ratio between the crank-radius and the length of the connecting-rod.

For any usual ratio $\frac{r}{l}$ the factor with which the crank-pin velocity must be multiplied to give the piston-velocity at the successive crank positions becomes:

$$\left(\sin\alpha + \tfrac{1}{2}\frac{r}{l}\sin 2\alpha\right)$$

		$\frac{r}{l} = \frac{1}{5}$	$\frac{r}{l} = \frac{1}{5.5}$	$\frac{r}{l} = \frac{1}{6}$
	360	0	0	0
30	330	0.587	0.579	0.572
60	300	0.954	0.946	0.939
90	270	1.000	1.000	1.000
120	240	0.78	0.788	0.794
150	210	0.413	0.421	0.428
180		0	0	0

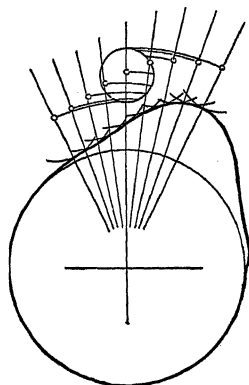
In any case when, for practical purposes, it becomes necessary to construct the piston-velocity curve, and the exact ratio $\frac{r}{l}$ is not definitely known, the intermediate values, for $\frac{r}{l} = \frac{1}{5.5}$, can safely be used, as the discrepancy, if any, is very unimportant.

It would, of course, be desirable to have the inlet- and exhaust-valves lift and seat uniformly with the increase and decrease in the piston-velocity, in which case the velocity of the gas through the valve-ports would be constant. This they would do, practically, if given a motion as if actuated by a crank which passes the centres in unison with the main crank.

The required form of the valve-cams, to give the correct crank-motion, would then be that shown by the construction, Fig. 91.

The valves do, however, never actually begin to open when the engine-crank passes the centres, and for practical reasons the valve-cams are generally given a simpler form than that shown in Fig. 91, and on these accounts the increase and decrease in effective valve-area does not follow uniformly the increase and decrease in piston-velocity. The curve, according to which the actual opening and closing of the valves normally occur in practice, is constructed in Fig. 90, in broken lines, and it will be noticed that it follows only approximately the rise and fall of the piston-velocity curve.

FIG. 91.

The curve representing the lifting of the valve is often, in practice, laid out in connection with the design of the valve-motion, in order to make sure that any improvement cannot be effected by a change in the form of the cams.

If the fluid velocity through the valve-ports be considered constant it is evident that the ratio between the maximum piston-velocity and the allowable maximum velocity of the gases in the valve-ports will determine the ratio between the full valve-area and the area of the cylinder. It is, however, more convenient, and more common, to determine the required valve-areas with reference to the mean piston-velocity, which is generally quoted, simply, as "the piston speed."

The latter is $P = \frac{2r}{\pi r} \times$ maximum piston-velocity

$$\frac{1}{1.57} \times \text{maximum piston-velocity.}$$

It is common practice to make the inlet- and exhaust-valves of sizes which call for fluid velocities through the valve-ports varying from 60 to 150 feet per second, counted with reference to the mean piston-velocity.

In small engines the valves can, without inconvenience, be made large, and they are to advantage often, in very small engines,

made as large as one-quarter of the cylinder-area. In large engines the valves and valve-springs become heavy, particularly when the valves are water-cooled, and for that reason the port-areas are cut down as much as can be done without incurring a considerable loss due to negative pressure on the piston. This it is particularly advisable to do with respect to large engines using cheap fuels as blast-furnace-gas or producer-gas, for the reason that a more reliable valve-mechanism will result. The inlet- and exhaust-valves are commonly made the same size. Sometimes, however, the exhaust-valve is reduced, so as to give an effective area only 80 per cent of the inlet-valve; this is done in order to reduce, as much as possible, the heavy strain in the valve mechanism required to open the exhaust-valve, against the pressure existing in the cylinder at the time for opening.

The following, Table XXX, gives the port-areas necessary, when it is desired to limit the fluid velocity in the ports to 60 feet, 100 feet, or 150 feet per second, as the case may be; the piston speeds being from 500 to 850 feet per minute. The last column of the table gives the required valve-diameter in terms of the diameter of the cylinder, with the assumption that the valve lifts enough so as to give the full area of the valve-port. This table is handy when proportioning port-areas to suit given conditions.

For instance, the table will show at a glance, that with a piston-speed of 500 feet per minute, and an assumed fluid velocity through the valve-ports of 100 feet per second, the required port-area becomes 8.33 per cent of the cylinder-area, or the valve-diameter 29 per cent of the cylinder-diameter. With a cylinder-diameter of, for instance, 6 inches, the valve would thus be only 1⅝ inches, which is very small. It will be better to limit the fluid velocity through the valve-ports to 60 feet per second, and make the valve-diameter 37 per cent of the cylinder diameter — 2¼ inches, or larger still if desired.

On the other hand, assuming the piston-speed to be 850 feet per minute, which is common in large engines, we obtain for a fluid velocity in the valve-ports of 100 feet per second a valve-area 14.2 per cent of the area of the cylinder, and a valve-diameter of 38 per cent of the cylinder-diameter. In the case of a 22-

inch cylinder, the valve would be 8⅜ inches, which is fully as large as it would be desirable to make it. As in this case the exhaust-valve, probably, would be water-cooled, a valve 34 per cent of the cylinder-diameter, calling for a fluid velocity of 140 feet per second, would, under most conditions, be preferable.

TABLE XXX.

Piston Speeds, Valve Port-Areas, and Gas-Velocities.

Piston Speed Ft. per min.	Vel. of Gases Through Valve-Port Ft. per sec.	Port Area in Per Cent of Cylinder Area.	Ratio Valve Diameter d to Cylinder Diameter D.
	60	13.9	$d=0.37D$
500	100	8.33	$d=0.29D$
	150	5.5	$d=0.23D$
	60	15.2	$d=0.39D$
550	100	9.2	$d=0.30D$
	150	6.1	$d=0.25D$
	60	16.6	$d=0.41D$
600	100	10.0	$d=0.32D$
	150	6.7	$d=0.26D$
	60	18.0	$d=0.42$
650	100	10.8	$d=0.33D$
	150	7.2	$d=0.27D$
	60	19.4	$d=0.44D$
700	100	11.5	$d=0.34D$
	150	7.7	$d=0.28D$
	60	20.8	$d=0.45D$
750	100	12.5	$d=0.35D$
	150	8.3	$d=0.29D$
	60	22.2	$d=0.47D$
800	100	13.3	$d=0.36D$
	150	8.8	$d=0.30D$
	60	23.6	$d=0.49D$
850	100	14.2	$d=0.38D$
	150	9.4	$d=0.31D$

Table XXX refers to mechanically operated valves, which are generally raised so as to give approximately the full opening of the valve-ports at its highest position.

The lift of a flat-seated valve corresponding to the full port-area is one-quarter of the valve-diameter, but, as inlet- and exhaust-valves are generally made with conical seats of approximately 45° angle, the lift should, theoretically, be somewhat more than that amount. However, the valve-seat being narrow, the area of the valve-stem may, in practice, be assumed to reduce the effective area to such an extent as to compensate for the discrepancy in the effective opening due to conical seat; and, in reality, the valves are seldom made to lift more than one-quarter of the valve-diameter.

The inertia of the valve, which the valve-cam has to overcome in opening it, and the spring in closing it, is, as will be shown, directly proportional with the weight and height of lift of the valve. But, the weight increasing in a much higher rate than in the direct proportion to the diameter, it would not serve the purpose of reducing inertia, to reduce the necessary lift of the valve by increasing its diameter.

The automatic spring-loaded valve, which can be lifted but a small amount, must be made of larger diameter than the mechanically operated one. The area of the former is ordinarily made 50 per cent larger than that of the latter, under otherwise similar service.

The extreme ratios between the valve-diameter, d, and the cylinder-diameter, D, will be found, in practice, to be:

For mechanically moved valves $d = 0.26\,D$ to $0.4\,D$,
and for automatic valves $d = 0.32\,D$ to $0.5\,D$.

The mechanically operated valve is generally made much heavier than what would be actually required for strength, in order to exclude any possibility of the valve warping or springing due to the heat to which it is exposed.

With respect to automatic valves, the necessary thickness of the valve-disc is suitably figured according to the formula for the

strength of round flat plates supported around the edge, which may be written:

$$t = 0.5\, d \sqrt{\frac{P_{max.}}{S}}, \quad . \quad . \quad . \quad . \quad . \quad (121)$$

t being the thickness required, in inches;
$P_{max.}$ the pressure per square inch of the valve-opening, and
S the strain per square inch of the material.

It must be remembered, in regard to automatic valves, that the durability of the valve, and valve-seat, will be increased by making the valve as light as the strength of the material will allow.

Assuming $P_{max.}$ to be 450 pounds, and allowing a strain $S = 12{,}000$ pounds per square inch, the thickness becomes

$$t = 0.09\, d.$$

This is, however, as light as a steel valve can safely be made.

The Valve-Seat.—The width of the face of the valve-seat should be made approximately 0.05 d.

Hence, for large engines the valve face becomes approximately $\frac{7}{16}$ inch, and for small engines the valve face becomes approximately $\frac{5}{32}$ inch.

The precaution is generally, taken with respect to more important engines, to make the exhaust valve-seat in a removable bushing which can readily be renewed in case of necessity.

The Valve-Stem.—The valve-stem serves to guide the valve properly, and to transmit the pressure that is required to cause it to open. The resistance against the opening of the inlet-valve is, principally, the tension of the valve-spring and a pressure due to the inertia of the valve. The latter force only is transmitted through the valve-stem. The resistance acting on the exhaust valve-stem is the inertia, in moving the value, and the pressure in the cylinder at the time it is opened. When the valve is being opened the inertia pressure is zero, and the pressure in the cylinder is, as a maximum, 60 pounds per square inch.

The valve-stem guide should be of ample length so as to guide and seat the valve properly, and, when so guided, the resistance on the valve-stem becomes, in effect, a simple compressive strain. This strain does not, ordinarily, need to be allowed higher than,

say, 3,000 pounds per square inch of the smallest section of the stem. Hence, if δ be the diameter of the smallest section,

we get
$$\delta^2 = \frac{60}{3,000} d^2,$$
or
$$\delta = 0.14 d.$$

δ is commonly the diameter at the bottom of the thread, in the end of the stem, and the main body of the stem will, of course, be increased a suitable amount for machining and fitting.

The Valve-Springs.—The valve-spring must fill two requirements: first, it must exert enough pressure on the valve, when closed, to prevent its opening on the suction-stroke; secondly, it must have the required force to close the valve promptly during the short time for closing.

The first requirement, which has reference principally to the exhaust-valve, calls for greatly differing spring-tension for different cases, depending on the manner in which the engine is governed. In the case of a throttling or cut-off engine, it is evident that the volume of the clearance-space will affect the amount of vacuum that can be obtained in the cylinder on the suction-stroke, when the engine is running light. In an engine having a clearance-space of, for instance, thirteen per cent of the total cylinder volume there would be obtained, possibly, a vacuum of 11 pounds below the atmosphere, whereas in an engine having a clearance of 25 per cent the vacuum would not be over 6 pounds below the atmosphere.

The weight of the exhaust-valve, which helps the spring to hold the valve closed, should, strictly, be deducted from the required spring-tension, but as ordinarily this weight is slight, generally not over $\frac{8}{10}$ of one pound per square inch valve-area, it may be neglected.

Accordingly, the exhaust valve-spring for throttling- or cut-off engines should have a tension, when in place, of 6 to 11 pounds per square inch valve-area; the latter figure referring to producer-gas or blast-furnace-gas engines with high compression.

In engines the regulation of which is effected by the admission of a proportionally increased volume of air at light loads (constant-

volume regulations), the vacuum never becomes as great as in the throttling engine, and hence the valve-springs can in such engines be made lighter.

Some hit-or-miss engines require exhaust springs of a tension of only 4 to 6 pounds per square inch valve-area.

Spring-Tension Required for Prompt Closing of the Valve.— In order to close the valve promptly, it is required that the spring shall have force enough to overcome the inertia due to the maximum change in velocity of the valve in closing.

Assume that a valve, weighing W pounds, is forced to its seat, a distance h inches, in T seconds, with uniformly increasing and decreasing velocity, such as the valve-cam would allow.

The maximum acceleration that must be given the valve is, approximately,

$$\text{acceleration} = 1.57 \frac{2}{T^2} \times \frac{h}{12} \text{ feet per second,}$$

and the force to give this acceleration will be

$$F = \frac{W}{g} \times \text{acceleration} = \frac{W}{32.2} \times \frac{1.57\,h}{6\,T^2} = \frac{W\,h}{123\,T^2}. \quad (122)$$

From Figs. 92 and 93, which are normal designs for an inlet and an exhaust valve-cam, it will be seen that the closing and opening throws extend, each, over a little less than ⅛ of the total circumference of the cam. In order that the cam-roller shall follow the closing cam-throw, it is necessary, therefore, that the valve closes in a little less than ¼ of one revolution of the engine. Assume, to be on the safe side, that the valve shall close in ⅙ of one revolution. Hence, the time for one revolution being $\frac{60}{N}$ seconds when N is the number of revolutions per minute, the time required for the closing of the valve will be

$$T = \tfrac{1}{6}\frac{60}{N} = \frac{10}{N},$$

and

$$T^2 = \frac{100}{N^2}$$

An ordinary mechanically moved valve (not water-cooled), weighs, approximately, 0.8 pounds per square inch valve-area.

Thus $W = 0.8 \times \frac{\pi d^2}{4} = 0.8\,a$, approximately,

and the lift of the valve is, ordinarily,

$$h = \tfrac{1}{4}\,d;$$

d being the diameter and a the area of the valve-opening.

Inserting the preceding approximate values of T_2, W and h, in equation 122, it becomes

$$F = \frac{d\,N^2}{61{,}500}\,a. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (122a)$$

This equation gives:

for $d = 8$ and $N = 200$, $F = 5.2\,a$;
and for $d = 3$ and $N = 400$, $F = 7.8\,a$;

which shows the probable limits for the value of F to be between 5 and 8 pounds per square inch valve-area. These pressures being less than the pressure required to hold the exhaust valve of high-compression throttling or cut-off engines closed during the suction-strokes at light loads, it is evident that, in the case of engines of the latter type, the exhaust valve-spring should, generally, be given a higher tension than that required for prompt closing—with exception in the case of high-speed engines with water-cooled valves, which require that the necessary spring-tension be ascertained by inserting the correct values of W, h and T in equation 122.

In the case of automatically operated inlet valves, which generally are made very light, a spring-tension of only ¾ to 1 pound will be required; the exact amount depending on the speed of the engine.

The Valve Cams.—When cams are used for actuating the opening and closing of the valves they should be given such a form as to strike and leave the roller in an easy manner. If this is done a noiseless valve-operation will be obtained. Figs. 92 and 93 show the commonly employed construction of the inlet- and exhaust-cams, with throws made of case-hardened steel. The cam-rollers, as is shown in the figures, clear, in their closed positions, the cams by a small amount. This clearance should

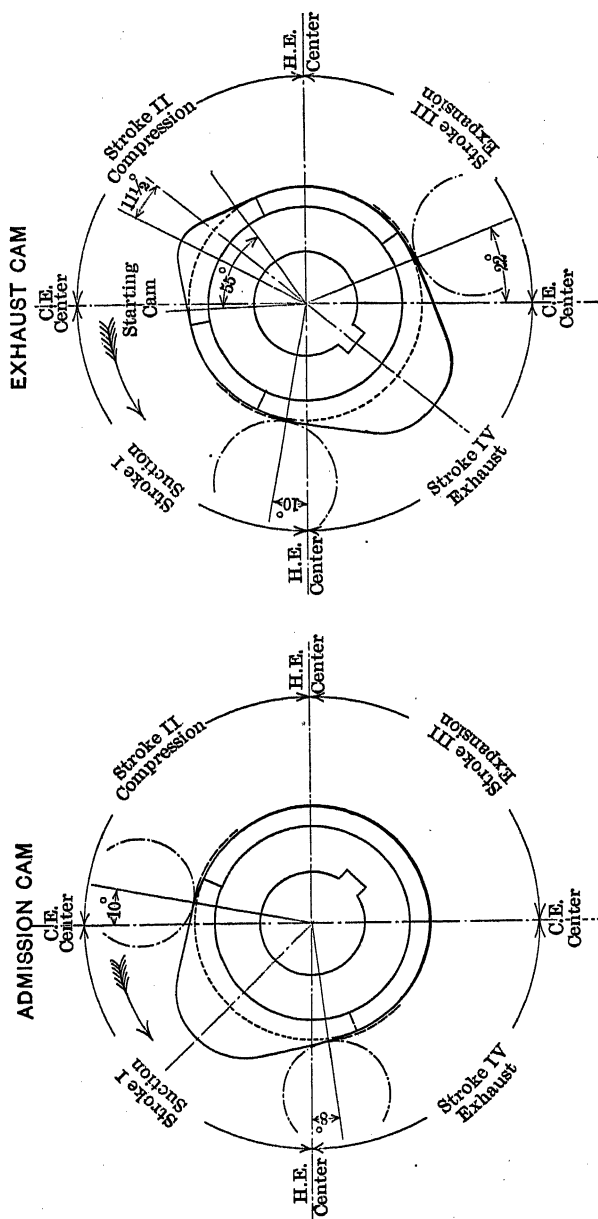



FIG. 92.

FIG. 93.

Diagrams of Valve-Setting for Producer-Gas Engines.

be carefully maintained after the valve-adjustment has been made, as a very small change will ordinarily throw the adjustment all out.

Valve-Setting.—The inlet-valve of a single-acting, four-cycle engine should properly open a little before the crank reaches the head-end centre, and close behind the crank-end centre. The exhaust-valve opens generally from 30 to 40 degrees before the crank reaches the crank-end centre, and closes 8 to 12 degrees behind the head-end centre. The exact setting of the valves must suit the speed of the engine and the nature of the fuel-gas used.

Figs. 92 and 93 are diagrams of the valve-setting used for producer-gas engines running at a speed of 800 feet per minute.

The Balancing of the Crank and Reciprocating Parts.—The simplest way to balance a revolving weight is by an equal weight placed on the opposite side of the shaft, and to balance, perfectly, reciprocating parts there must be used similar parts moving symmetrically in an opposite direction to those to be balanced. Vertically reciprocating parts can be perfectly balanced vertically, and horizontally reciprocating parts may be perfectly balanced horizontally by suitable revolving masses moving in an opposite direction to the reciprocating parts. In balancing reciprocating masses by revolving ones there will always, however, be evolved new unbalanced forces acting perpendicularly to the direction of motion of the reciprocating parts. Thus, in balancing vertically reciprocating parts there will result horizontal unbalanced forces, and in balancing horizontally reciprocating parts there will be evolved vertically unbalanced forces.

In balancing reciprocating parts it is of advantage to use a light weight placed as far as possible away from the centre of the axis, and when two wheels are used one-half of the total balance-weight may suitably be put at the periphery of each wheel. The whole balancing weight cannot be put in one wheel, excepting by locating another suitable weight on the opposite side of the centre-line of the engine. If this second balance-weight is not employed there will result a rocking force not normal to the centre-line of the engine.

In conformity with these principal facts regarding the balancing of engine-parts, it is customary, in an upright engine, to balance the revolving parts only, the crank-arms, the crank-pin and one-half the weight of the connecting-rod. A complete balancing of the reciprocating parts is not attempted, as it is better to have a free unbalanced force vertically, in which direction it can be resisted more effectively, than to change it into a horizontal unbalanced force.

In a horizontal one-wheel engine seven-tenths of the weight of both revolving and reciprocating parts are generally balanced.

Accordingly, if W_c is the total weight of the counter-weight;
r_c the radius at which it is applied, in inches;
r the radius of the crank, in inches;
W_1 the weight of the revolving masses, including the unbalanced weight of the crank-arms reduced to the radus r, the crank-pin and one-half the weight of the connecting-rod;
W_2 the weight of the reciprocating parts, including the weight of the piston and pin, and one-half the weight of the connecting-rod;

Then, for an upright engine

$$W_c = \frac{r}{r_c} W_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad (123)$$

and for a horizontal engine

$$W_c = 0.7 \frac{r}{r_c} (W_1 + W_2). \quad . \quad . \quad . \quad . \quad (123a)$$

In large tandem engines it is often impossible to apply to the crank the full required balance-weight, and it becomes of advantage to apply part of the weight to the crank and part to the fly-wheel rim.

When two balance weights, on different distances from the centre of the engine, are employed, their relative weights are determined as follows:

In Fig. 94, W_1 is the weight to be balanced, acting at a radius r_1.

W_2 and W_3 are the balance-weights, located at distances l_2 and l_3 from the centre of the engine, and acting at radii r_2 and r_3.

The centrifugal force due to a weight W acting at a radius r inches, at N revolutions per minute, is

$$C = 0.0000285\ N^2 r W.$$

In order to obtain balance between the forces C_1, C_2 and C_3 it will be required, first, that $C_1 = C_2 + C_3$;

thus, $$W_1 r_1 = W_2 r_2 + W_3 r_3, \quad . \quad . \quad . \quad . \quad (124)$$

and, secondly, that the moments relatively to O are zero;

thus, $$W_2 r_2 l_2 = W_3 r_3 l_3. \quad . \quad . \quad . \quad . \quad (125)$$

Hence, if the radii r_2 and r_3 and the distances l_2 and l_3 be

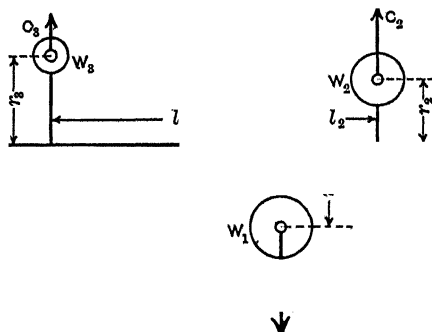



FIG. 94.

established, the weights W_2 and W_3 can readily be determined from equations 124 and 125.

Counter-Weight Bolts.—The bolts holding the counter-weight to the crank must be figured amply strong to resist the centrifugal force acting on it. This force is

$$C = \frac{12\, W_c V^2}{g\, r_c} = 0.0000285\ N^2 r_c W_c. \quad . \quad . \quad (126)$$

W_c being the weight of the counter-weight,

r_c the distance from its centre of gravity to the axis of rotation in inches;

V the linear speed of the centre of gravity, in feet per second; and
N the number of revolutions.
g is the acceleration due to gravity $=32.16$.

Fig. 95 illustrates a common and reliable way to secure the

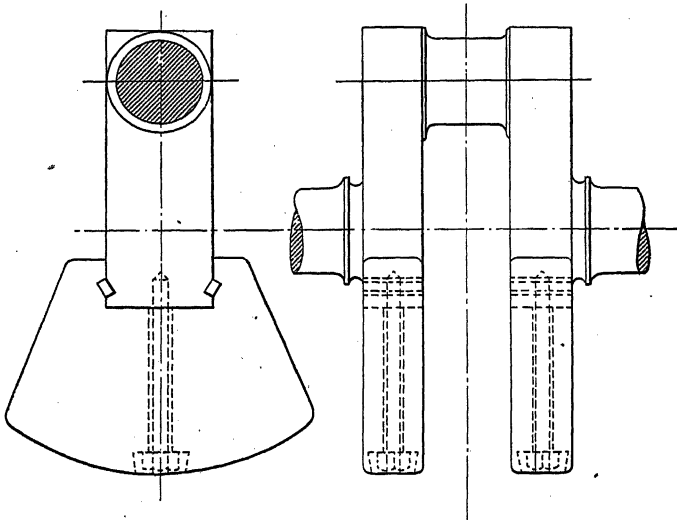



FIG. 95.

balance-weights to the crank-arms. The dovetailed nut pockets are, after the bolts are tightly in place, filled with lead to insure against the bolts unscrewing.

Water Cooling.—All gas-engines, excepting the very smallest,

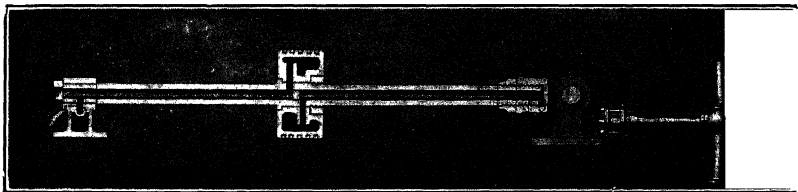



FIG. 96.—Water-Cooled Piston.

which may be self-cooling, must be provided with some means for cooling the cylinder. Cylinders of a size up to 4 inches may be cooled by an air-current which is made to impinge on the

cylinder-jacket, preferably, then, provided with a ribbed cooling-surface to facilitate the dissipation of heat.

Cylinders over 4 inches are always water-cooled. The larger the cylinder, the more important it is to thoroughly cool the combustion-chamber, particularly around the exhaust valve-seat and spark-plugs. Faulty arrangements for the cooling of these parts, which are most liable to become over-heated on heavy loads, will cause pre-ignitions for compressions that otherwise would be safe and normal for the particular fuel used.

FIG. 97.—Water-cooled Exhaust-Valve

Single-acting engines above 20 to 24 inches, and double-acting cylinders generally, are provided with water-cooled pistons and exhaust-valves; and the more liable the fuel is to cause pre-ignitions the more necessary it will be to cool these parts.

Fig. 96 shows a common method for cooling the piston and rod of double-acting engines. The cooling water is, by means of a telescoping tube, admitted to the hollow piston-rod, at the main cross-head, and it passes from there to the bottom part of the piston. As it becomes heated the water rises to the top of the piston and is discharged from the very highest point of it so as to, as thoroughly as possible, expel with it any air or steam that may tend to separate off. The hot water is drained from the rear cross-head, in a visible stream, by which its temperature and proper continuance can always be observed.

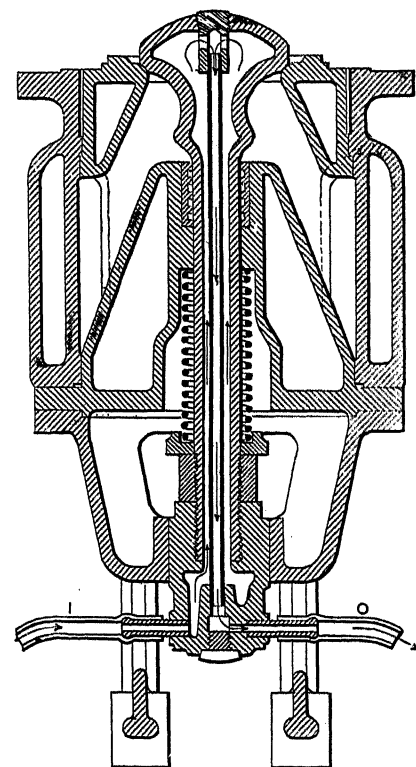



Fig. 97 illustrates a water-cooled exhaust-valve of modern type. The cooling water is admitted through a flexible hose, at *I*, and passes through the hollow valve-stem to the crown of the valve, from where it is discharged through a small central tube which stands in communication with the discharge hose *O*. From the discharge hose the water is drained, openly, into a drain funnel so that the flow can always be observed.

Cooling Water Required.—It may be stated, roughly, that as a maximum one and one-half times to twice as much heat is carried off by the cooling water as that converted into work. Accordingly, each horse-power corresponding to 2,545 heat-units per hour, the cooling water absorbs

$$(1\tfrac{1}{2} \text{ to } 2) \times 2{,}545 \text{ B.T.U. per hour per horse-power,}$$

or from 3,817 to 5,090 B.T.U. per hour per horse-power.

Each pound of the cooling water absorbs

$$(t_f - t_i) \text{ B.T.U.,}$$

when t_f and t_i are the temperature-limits between which it is heated; and, as the temperature of the jacket-discharge should ordinarily not be allowed to become over 185° F., it may be said, that approximately from 100 to 150 B.T.U. are absorbed per hour per pound of cooling-water.

Hence, the quantity of cooling-water ordinarily required per horse-power, will be between

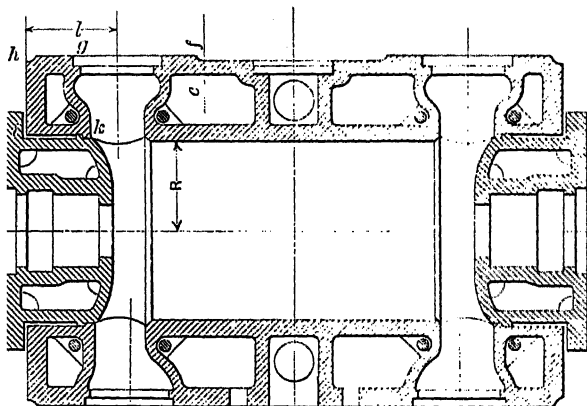
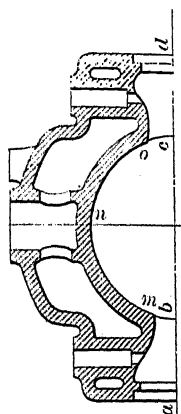
$$\frac{3{,}817}{150} \text{ and } \frac{5{,}090}{100};$$

from 25 to 51 pounds per hour,
or from 3.1 to 6.3 gallons per hour.

Allowing for any ordinary overload-capacity of the engine, a circulating pump supplying 8 gallons per each rated horse-power would be ample, provided the water is not supplied hotter than 85° F. For each 10° of temperature higher than 85°, at which the water is supplied, 10 per cent more pump-capacity will be required. It will be noticed that, if these requirements are filled, the pump-capacity will, ordinarily, be approximately 50 per cent above actual requirements.

The Double-Acting Cylinder.—A double-acting cylinder, as frequently carried out for the four-cycle engine-type, is shown in Figs. 98 and 99. The strains in the material of a cylinder of this construction, which not infrequently have caused its failure, are due to two separate causes: Strains due to the working pressure, and strains due to the expansion of the inside cylinder-wall, when heated by the working gases.

It is readily ascertained, that the part of the cylinder-barrel between the openings cut through it for the accommodation of the valves will without difficulty be made amply strong for sus-

FIG. 98. FIG. 99.

Nuernberg Double-acting Cylinder.

taining the tangential strains due to the explosion pressure; figured as a plain cylindrical shell carrying internal pressure. Whether the axial strains, in a direction lengthwise of the cylinder, will be taken up by the inside barrel, or by the outside jacket, depends on the arrangement made to allow for the expansion of the inside wall.

The design of the cylinder illustrated is that used by the firm The Maschinen Gesellschaft Nuernberg, Germany, and its weak sections have, in practice, been found to be in the sections *a b* and *c d*, and also in the outside jacket, at section *e f*. Due to the openings for the valve-bonnets, which cut through the outside

and inside walls, it becomes necessary that the section *g h i k* is of ample strength to sustain a total load $P_{max.}\ l\ R$, by which it will be strained, and to obtain strength the distance *h i* must be made of a considerable depth, or steel stay-bolts, as shown in the figure, may be used to help carry the load. Further, as the abutments for the arched part *m n o* are cut away by the openings for the valve-bonnets, there will be caused transverse strains in the sections *a b* and *c d*, that must be taken in consideration.

When the cylinder is made, as in the illustration, with the outside jacket-wall cast continuous from end to end, the question will arise, what will be the effect of the strains caused by the expansion of the material on the strength of the construction as a whole.

It is reasonable to assume the temperature of the inside surface of the cylinder-barrel to be, on an average, 500° F., the cooling water 100°, and, hence, the average temperature-difference between the inside barrel and the outside jacket, approximately, 200° F. Such a temperature-difference would call for an expansion of the inside barrel of practically one-sixteenth of one inch, or more, relatively to the outside jacket-wall. The elasticity of the material will allow of this, assuming the water-space is made of proper depth. The Nuernberg construction is given adequate depth of water-space, but there will result in certain sections of the jacket a not inconsiderable tensile strain; and the inside cylinder-barrel will be, axially, in compression. The axial strains, therefore, due to the pressure on the piston will all be taken by the cylinder jacket, which, consequently, must be made of proper thickness and so ribbed that the axial strains become evenly distributed.

Some builders prefer to have the jacket core cut through the outside cylinder-wall at the middle of the cylinder, to avoid any longitudinal strain due to the expansion of the metal by the heat transmitted from the working gases. The cylinder-barrel will, then, take all the axial strain, which it fully can do, without increased stress, because the strain axially is only one-half as severe as that tangentially. The latter construction involves some little difficulty in making a water-tight expansion-joint all

around the cylinder, and this is what has been sought to be avoided in the Nuernberg construction.

The Piston-Rod Packing.—The main idea on which the construction of all modern rod packings is based is to reduce gradually, and as effectively as possible, from cell to cell of the packing, the pressure existing in the cylinder to that of the atmosphere; taking in consideration that ample provision must be afforded the rod to centre itself freely between its points of support at the

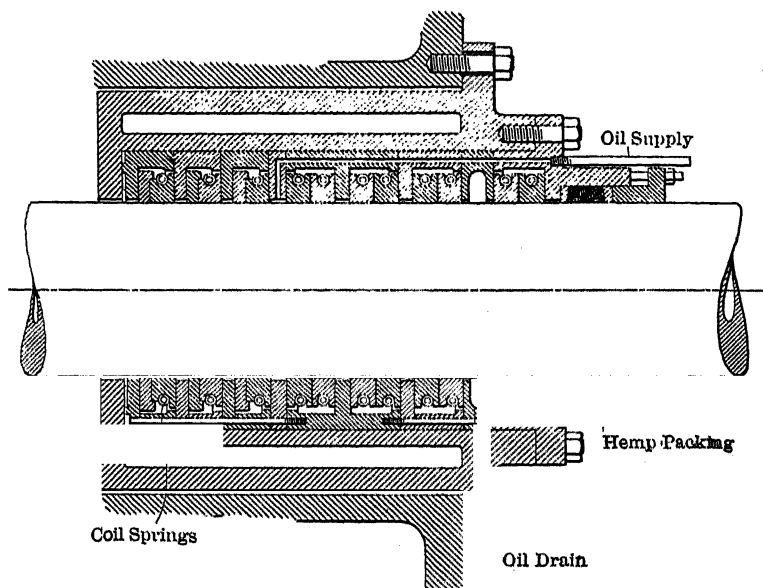



FIG. 100.—Schwabe Piston-Rod Packing.

main and at the rear crosshead. Of the construction of the rod-packings generally employed the Schwabe packing, Fig. 100, may serve as an illustration.

It consists of several pairs of three-parted soft cast-iron rings, breaking joints; the ring-sections being held together and sprung against the rod by suitable coil-springs. The packing is self-contained in a casing that can readily be taken apart, for the removal or examination of the rings.

An effective lubrication and circulation of the lubricant is brought about by supplying the oil at a point from where it will become well distributed over the rings without tending to leak too freely into the cylinder, and by draining it from the front end of the packing. To prevent the lubricant from escaping along the rod, a soft hemp-packing is provided outside of the last set of rings. When the rod is water-cooled it will generally not be necessary, sometimes not even desirable, to cool the packing.

A modified form of piston-rod packing is illustrated in Fig. 100*a*. The number of rings usually employed in this packing

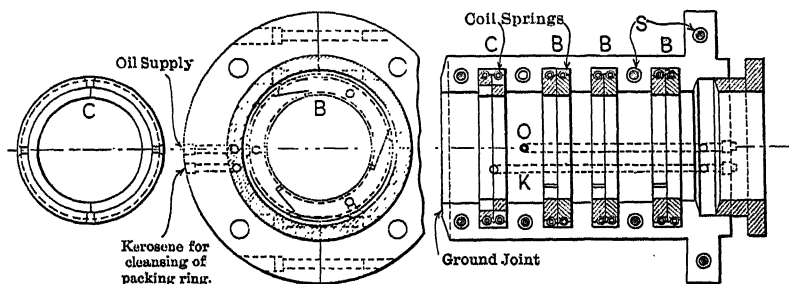



FIG. 100 *a*.—American Piston-Rod Packing.

varies somewhat; four or five sets being ordinarily used. The set of rings most liable to become impaired on account of impurities, tar, or deficient lubrication is, of course, the inside set, and these rings, besides, have usually to bear a heavy pressure due to the gas leaking past them into the first groove. Instead of using self-adjusting rings in the first groove these rings may be fitted to the rod in halves with the ends abutting, as shown at *C*. As the casing is made in halves held together by the small screws *S*, the rings are readily gotten at for re-fitting, when necessary. *O* is an oil-hole and *K* a channel through which kerosene may be forced into the first groove for the purpose of cleansing the rings.

CHAPTER XII

GOVERNING

THE methods generally employed for controlling the speed of a gas-engine are: either the so-called hit-or-miss method or the throttling or cut-off method, or that of advancing or retarding the ignition.

Hit-or-Miss Governing.—In the first system the charge is cut out entirely for one or more pressure-strokes, whenever the speed becomes above the normal.

This is accomplished by the governor, by withdrawing a member, a pick-blade or a cam-roller, in the valve-actuating mechanism, causing the inlet-valve to remain closed against the admission of new charge. In case the inlet-valve is operated automatically by the suction of the piston, the governing may be actuated upon the exhaust-valve by holding it open during the admission-stroke, thereby preventing the spring-loaded inlet-valve from opening.

A great variety of hit-or-miss governing devices are in use, mostly in connection with small-sized engines where any particularly close regulation is not very essential. The governor proper may be a fly-ball governor or one of the inertia or pendulum type.

Various Forms of Hit-or-Miss Regulations.—Fig. 101 illustrates a form of hit-or-miss regulation controlled by a fly-ball governor, in combination with a pick-blade.

S is the cam-shaft, which generally is driven from the main engine-shaft by means of a pair of spiral gears, in such a manner that it makes one revolution while the main shaft makes two. The cam *C* secured to the cam-shaft, may thus be timed so as to actuate the cam-roller *D*, which is carried by the valve rocker *R*, at the proper time for the opening of the inlet valve *I*. The position of the pick-blade *P* is controlled by the governor, by means of the bell-crank *B*, so that, when the governor is running

below normal speed, the pick-blade is carried in a position to engage with the valve-stem, and open the valve; but when the speed is above the normal, the pick-blade will be in such a position as to miss the valve-stem, and the valve will remain closed.

Fig. 102 illustrates an inertia governor which regulates on the hit-or-miss principle by means of a pick-blade. It is a type used by the firm Crossley Bros., Ltd., Manchester, England, on small four-cycle engines. *S* is the cam-shaft and *C* the cam which

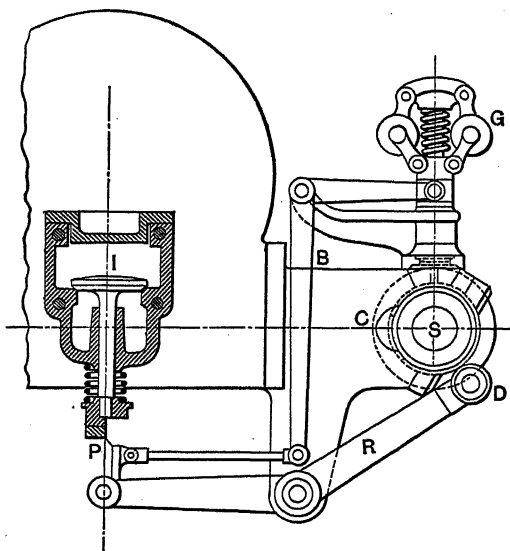



FIG. 101.

actuates the cam-roller *D* at the proper time for opening the inlet-valve. *A* is the governor arm, which on one end carries the weight *G*, and on the other the pick-blade *P*. The valve-rocker *R*, which transmits the motion from the roller *D* to the valve stem *I*, is fulcrumed at *F*, and carries the governor arm hinged on it at *H*. The governor arm, with the weight and pick-blade, is thus free to swing through a small arc relatively to the rocker *R*, excepting that the spring *M* holds it with a suitable force against the rocker, so as to permit the pick-blade, under certain circumstances, to meet the end of the valve-stem *I*, when the rocker is actuated

upon by the cam *C*. *K* is the valve-spring which holds the valve closed, and the spring *L* has for object to push the rocker *R* back to its neutral position.

The general motion of the governor-weight is along the arc $a - b$ whenever the tension of the spring *M* is strong enough to hold it in this path. At a certain speed of the engine, however, the cam *C* will hit the roller *D* too swiftly for the spring to hold the weight in its regular path, and the inertia of the weight will then throw it back and carry the end of the pick-blade downward

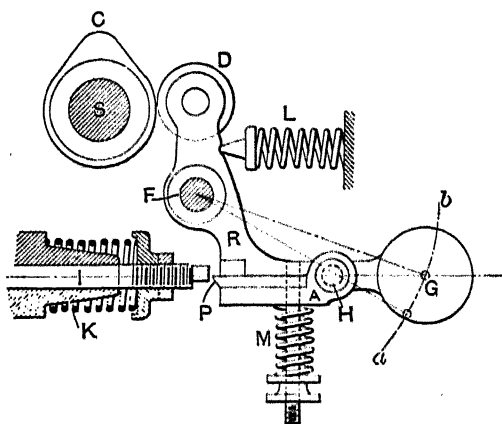



FIG. 102.

so as to miss the end of the valve-stem. When this happens the valve will remain closed against the admission of any charge.

Fig. 103 illustrates a so-called pendulum governor which occasionally is used for the regulation of small engines. *S* is the cam-shaft and *C* the cam which actuates the pushrod *P*. The latter is guided at *D* and *E*, and is carried back to its neutral position by the spring *L*. On a fulcrum-bracket, *B*, secured to the pushrod there is hinged a pendulum, *G*, which is made in one piece with the pick-blade arm *A*. *I* is the inlet valve-stem and *K* the valve-spring.

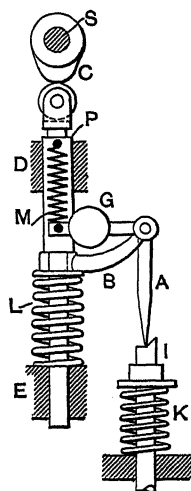
When the pushrod is given a reciprocating motion, in time with the required periods for opening and closing the valve, the pendulum will be set in a swinging motion, which at excessive

speeds will have for effect to carry the pick-blade out of contact with the end of the valve-stem and thus cause one or more miss-strokes. The spring *M* has for object to steady the pendulum and to reduce the arc of its oscillations, which, without it, would be excessive.

In Fig. 104 there are shown various arrangements of the pendulum-governor. In the arrangements *I* and *II* there are no retarding springs used, and as a consequence the swing of the pendulum and the sensitiveness of the regulation will be excessive. The arrangement *III* is practically the same as that of Fig. 103, and in the arrangement *IV* the valve will be hooked in and follow the motion of the push-rod, until, at excessive speeds, the mechanism becomes unhooked, leaving the valve to remain closed for one or more strokes.

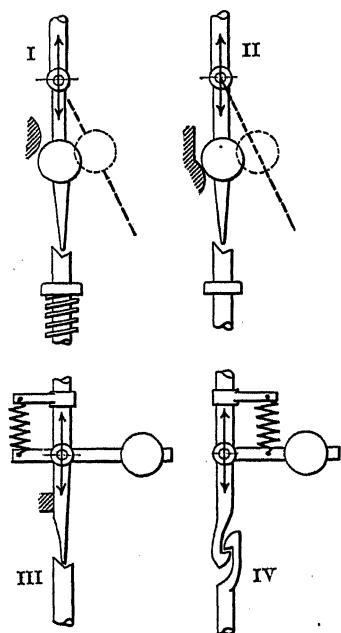
FIG. 103.

The Throttling or Cut-off Regulation.—In the throttling or cut-off regulating system the explosions and pressure-strokes follow each other always at regular intervals, but with diminished intensity at light loads, due to the admission of a charge of impoverished fuel-value. This system of governing, which is often applied to modern well-built engines, may be carried out according to two different methods. By the first, the so-called *constant-quality* method, the charge remains for all loads of a constant proportion of air and fuel, but at light loads it will be throttled, or cut off, during the suction-stroke to suit the load. The density of the charge at the end of the suction-stroke, as well as, consequently, its compression becomes, therefore, variable, and proportionate with the load.

By the second, the so-called *constant-quantity* method, the quantity of the charge remains uniform for all loads, but its quality is made poorer at light loads by means of the throttling or cutting off of the fuel alone; the air charge, strictly, must be increased by the amount the fuel is decreased, or, when the fuel is rich, it remains practically constant. As the density of the

charge after completed suction-stroke remains constant for all loads, therefore, the compression of the charge will also be constant under all load conditions.

When the fuel-charge alone is regulated by a cut-off valve, the object should be to admit the fuel as late during the suction-stroke as practicable, by which the advantage is sought, to obtain, as far as possible, a charge consisting principally of air near the piston, and which gradually grows richer in fuel nearer the igniter. For this purpose, the fuel-valve is opened late and closed near the end of the suction-stroke.

FIG. 104.

Constant-Quality Regulations.—The regulation system of the engine illustrated in cross-section in Fig. 105, consists of a butterfly valve *V*, controlled by the governor, by means of the governor-shaft *S*, the lever *L*, and the connection *C*. The gas arrives through the nozzle *N*, and the air through the opening *A*, into the mixing-chamber *M*. When the governor rises it closes the butterfly-valve more or less; thus, by throttling, it reduces the density of the charge so as to admit to the cylinder heating-valve in proportion to the momentary load requirements. This is a constant-quality throttling governing.

Instead of the governor throttling a charge of a constant mixture of gas and air by means of a butterfly-valve, a cut-off valve is sometimes used, which, under the control of the governor, cuts off the charge at a point of the stroke suitable to the load, similarly as in the steam-engine. An example of this class of governing is that of the Jacobson engine illustrated in Fig. 147, page 373.

Another type of constant-quality regulation is that employed by the Gasmotoren Fabrik Deutz, Cologne, Germany, and which is illustrated in Fig. 106. In this regulation the gas and air arrive to the admission valve through separate channels in which the gases are throttled to the correct proportions, and they are, at reduced loads, wire-drawn in the gas and air valve-ports, to the

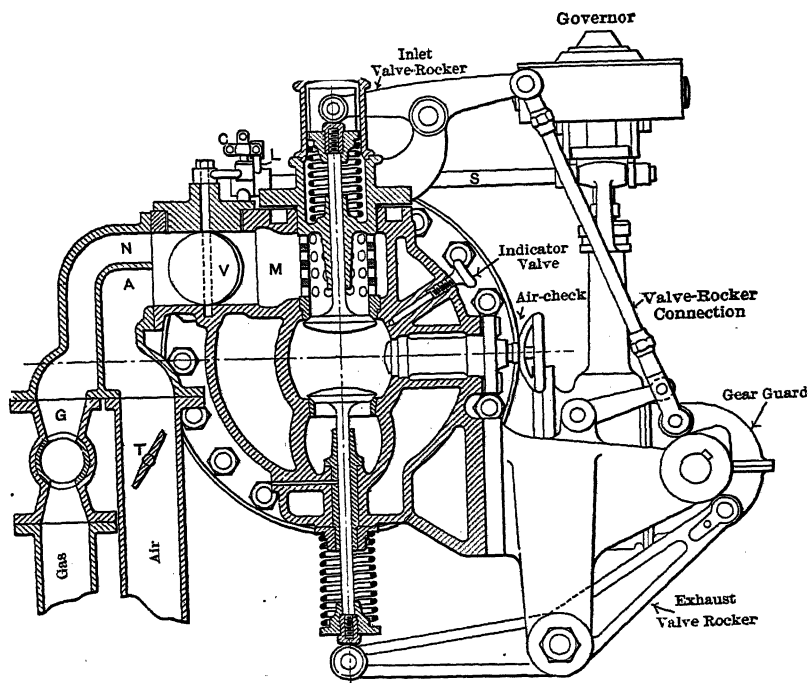



FIG. 105.

extent necessary for reducing the density of the mixture to correspond to the momentary load requirements.

Referring to the figure it will be seen that the air- and gas-valves, which are made in one piece, are slipped over the main inlet valve-spindle and held there between suitable collars. *R* is the admission cam-roller, which is actuated upon by the cam *C*, and which transmits motion to the valve-lever *L* by means of the valve-rod *D*. The valve-lever is not hinged to any fixed fulcrum, but a

movable roller, *M*, serves as fulcrum for depressing the admission valves. The position of the fulcrum-arm *A* is, as seen, controlled by the governor. For the position of the arm as shown in the figure the valves will obtain their maximum lift, but for a rising governor the fulcrum-roller *M* will move toward its inward position, at *E*, and thus, successively, increase the leverage with which the valves are actuated upon and cause their lift to be decreased. As the port-areas for the gas and air will always

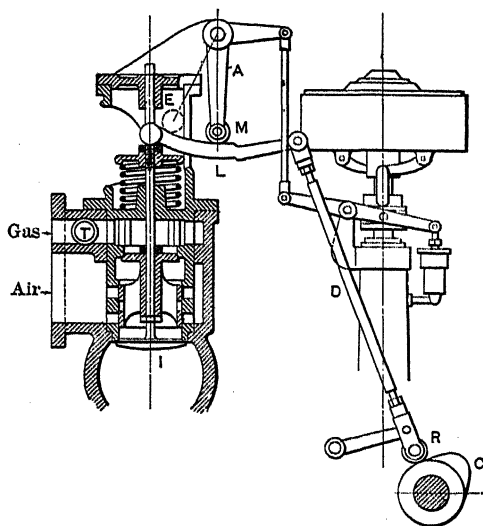



FIG. 106.—Deutz Admission Valve-Gear.

increase and decrease proportionately, therefore, the proportion in which the gas and air is admitted will remain constant.

T is a throttle in the gas-pipe by which the permanent proportioning of the gas and air is controlled.

If the charge actually remains of a constant quality by the use of the throttling or by the use of the cut-off regulations, there is then very little difference between the results derived by the two systems, because the charge must in either case be partially, or completely, excluded at one time or other during the admission-stroke, to the extent that the same quantity of charge is admitted; and the compression becomes in either case the same. Practically

there is, however, some difference between the two methods, because by the employment of a cut-off valve the quantity as well as the quality of the charge may be varied.

Constant-Quantity Regulations.—To obtain a constant-quantity regulation the gas must be throttled separately to suit the variations in the load. Such a regulation is illustrated in Fig. 107. The inlet-valve opens in this regulation under all load-conditions to its full lift, admitting always the full volume of charge, and while the gas may be throttled more or less the air

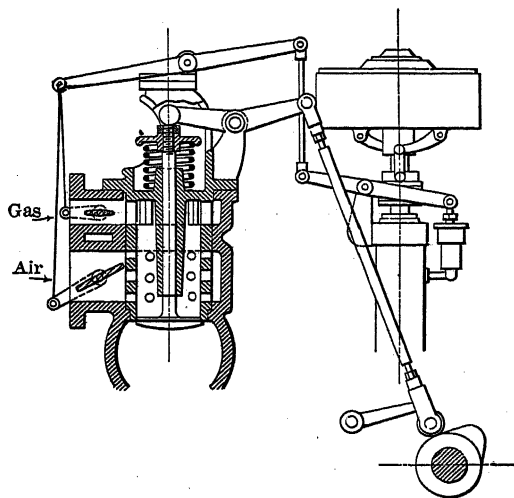



FIG. 107.—Constant-Quantity Regulation

charge is in a corresponding proportion increased. That is, when the gas-throttle is closed the air-throttle is correspondingly opened; both valves being under the control of the governor.

Fig. 161, page 397, represents a constant-quantity regulation in which the fuel-charge is regulated by means of the gas-valve, which is during the admission-stroke opened to a greater or less extent; the valve closing regularly at the end of the suction-stroke.

It is well known, and it will be shown presently by the results obtained from indicator cards taken at full and light loads, that the economy of the throttling or cut-off engine decreases rapidly

as the load decreases. The economy of the hit-or-miss engine, on the other hand, is more constant for all loads; as would be expected when all pressure-strokes are made under much the same conditions as to compression and mixture. The only dissimilarity between one power-stroke and another is in this engine due to the fact that after one or more misstrokes the cylinder becomes more thoroughly scavenged from the burned gases, and, perhaps, cooled by the air which is taken in and expelled during the misstroke.

In order, therefore, to attain as high economy as possible, under all load-conditions, the Crossley Bros., Ltd., Manchester, Eng., have, for years, been using a combination throttling and hit-or-miss regulation for engines running on variable loads. This regulation has also been made a feature of the Crossley engine manufactured in this country.

A new combination-regulation of this type was recently described by Mr. Atkinson, in a paper read before the Institution of Mechanical Engineers of London.

The regulation is shown in Figs. 108, 108*a* and 108*b*, and its principal feature is that, for loads above one-half the rated load of the engine, it acts on the throttling governing principle, whereas for loads below one-half the rated load it acts on the hit-or-miss principle. The latter feature is readily observed by a glance at the illustration. When, namely, the governor arm *L*, due to a light load, rises above a certain limit, it pulls the block *B* away from the path of the pick-blade, *P*, which is secured to the valve-rocker, *R*, and, as a consequence, the pick-blade passes below the block, thus failing to open the valve.

The inlet valve-stem is made hollow, and it is connected with a dash-pot plunger, *D*, moving air-tight in the vacuum-chamber, *C*. *S* is the valve-spring which holds the inlet-valve closed, and, the plunger and the valve being in one piece, the same spring serves also to push the plunger to the bottom of the vacuum-chamber. To allow the plunger to move to the bottom of the vacuum-chamber and close the valve promptly, there is, at *V*, a small snifting-valve opening outwards.

The force that opens the main valve when the pick-blade

engages with the block *B* comes through a stiff spring, *T*, from the push-head, *H*, which, normally, butts against a shoulder at the outside end of the hollow valve-stem; the spring *T*, thus, merely serving as a flexible valve-stem for opening the valve.

At *A* is a small air valve, which throttles the air-port leading to the vacuum-chamber, *C*. This valve is, by means of the connection *K*, controlled by the governor. When the governor-

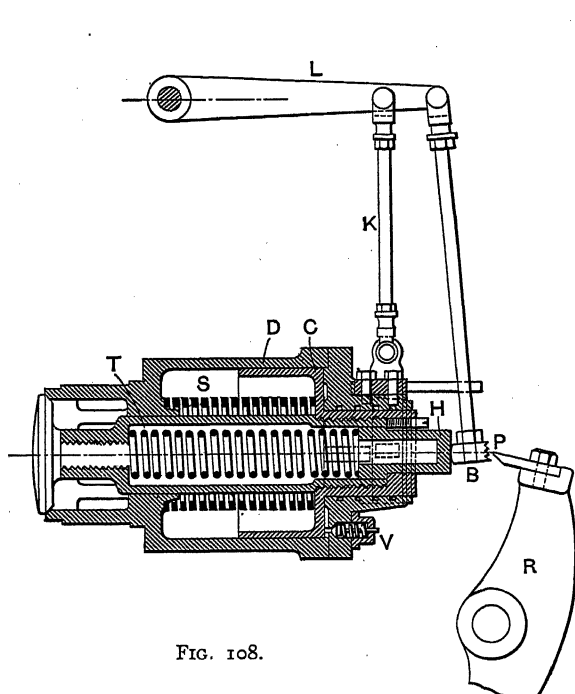



FIG. 108.

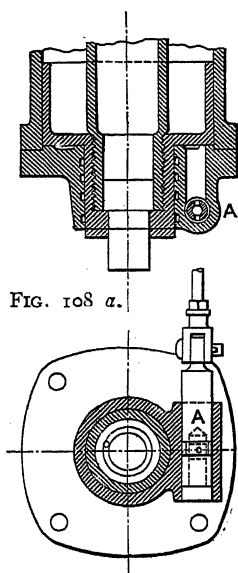



FIG. 108 *a*.

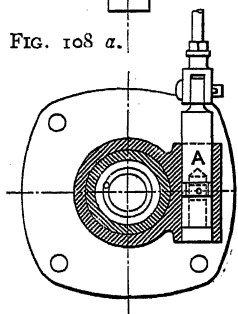



FIG. 108 *b*

arm swings low, at heavy loads, the valve *A* will open the air-port fully, thus preventing any vacuum from forming back of the plunger, *D*, and, hence, the inlet-valve will move freely to its full lift and admit maximum charge. When, however, the governor-arm swings high, at light loads, then the valve *A* will throttle the air-port leading to the vacuum-chamber, more or less, throwing a proportionally greater strain on the spring, *T*, in forcing the main valve open. This will, of course, have for effect to compress

the spring T, more or less, and to reduce the lift of the valve correspondingly.

Regulation by Retarding the Ignition.—In very small engines, temporary changes in the load may be taken care of without disturbing the regular working of the valves or altering the quality of the charge, simply by retarding the spark when a reduced effect is required. This system is of course wasteful, as the same quantity of fuel will be consumed at light as at heavy loads, and it is used only when the actual efficiency of the engine is of minor importance.

The Governor.—The Hartung type of governor is, at present, very commonly used for the regulation of gas-engines of medium and large size. This governor, illustrated in Figs. 109 and 110, is of fly-ball type, using springs for counter-acting the centrifugal force of the weights, and it has very generally been adopted on account of its sensitiveness, minimum amount of internal friction and great power; it being one of the types that for the least weight of the fly-weights possess the greatest power. Another feature of this governor, that recommends itself favorably, is that the fly-weights can conveniently be enclosed in a casing, which makes attending to the governor, when in motion, entirely safe, even in a crowded space. The governor is generally driven from the cam-shaft of the engine by means of a pair of spiral gears, or, occasionally, by means of spur- or bevel-gears. The spiral driving-gear, G, in the illustration Fig. 109, is, it will be seen, free to slide up on the driving-sleeve a small distance; excepting for the pressure by which it is held down by the spring S. The object of this arrangement is to give elasticity to the sudden starting of the governor.

Rite's governor is occasionally used for regulating the distribution of the charge in the gas-engine. The valves are then actuated by means of an eccentric, the position of which is under control of the governor.

In connection with large multiple-cylinder engines where several fuel-valves are to be regulated by the governor, its power may become insufficient to handle the regulation, directly. In such cases an indirect regulation may be effected by means of a

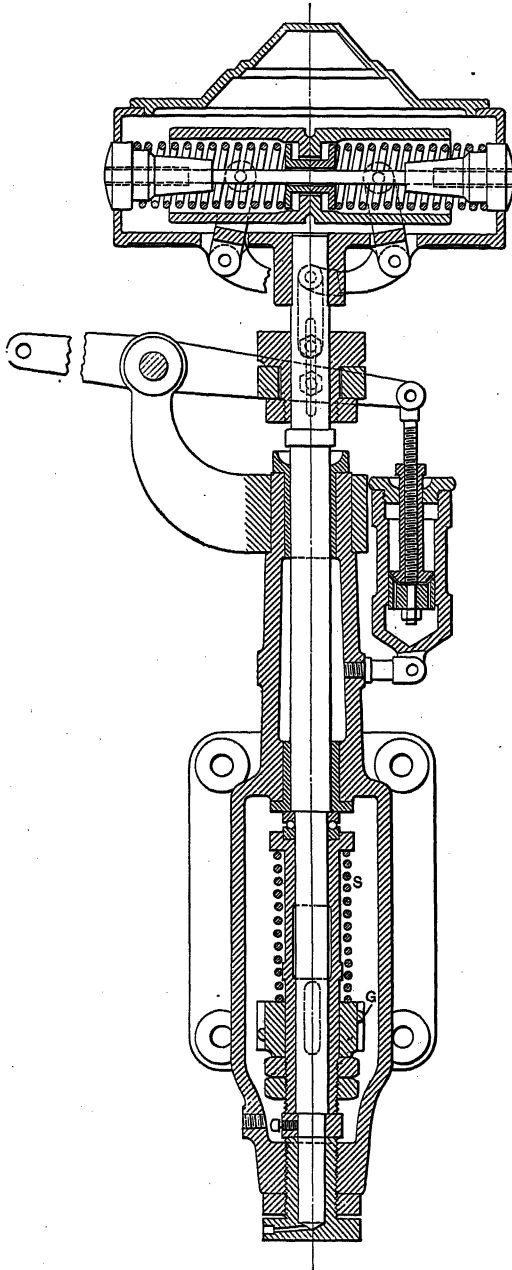



FIG. 109.

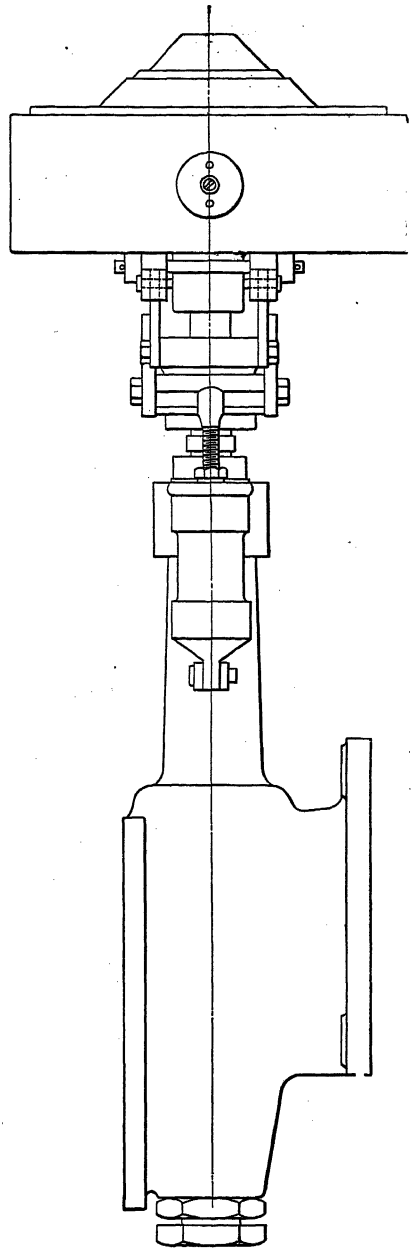
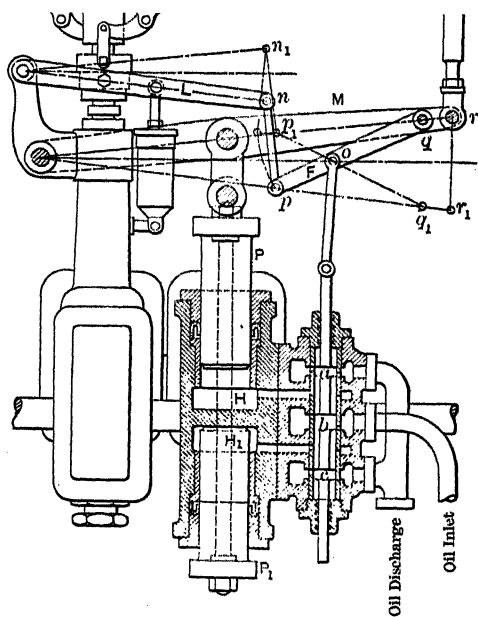



FIG. 110.

hydraulic piston, the motion of which is controlled by the governor. Fig. 111 represents, diagrammatically, an arrangement for such an indirect regulation that has occasionally been used for the governing of large engines.

Referring to the figure, P, P_1 are outside packed plungers actuating the controlling lever, M, which regulates the engine valve-gear. The governor is shown in its low position—the

Fig. 111.

governor-lever L being down, and the hydraulic plungers held up by the pressure acting in the top chamber of the hydraulic cylinder.

If the governor rises, say, to its top position, it will lift the floating lever pin p to p_1, and also lift the hydraulic controlling-valve, which consists of the three spools a, b and c having a sliding

fit in the valve-chamber sleeve. The oil, under pressure, will then be admitted below the middle spool, *b*, through small holes drilled in the sleeve, and enter the lower hydraulic chamber, H_1. At the same time there will be effected an outlet for the oil from the top chamber, *H*, when the top spool, *a*, rising, uncovers the outlet port of the controlling-valve. The hydraulic plungers will, thus, descend, bringing the lever *M* down, until the pin *q* comes to q_1. The floating lever will then have the position p_1 o q_1, and will thus have brought the controlling-valve back to its normal position for closing the oil-admission and discharge.

Accordingly, whatever the position to which the governor-lever *L* may be brought, the hydraulic plungers will instantly bring the lever *M* to a corresponding position and hold it there, until further called upon by the governor to again regulate its position.

Advantages of the Different Regulating Systems.—The factors that determine the output of an engine are: The amount of gas admitted, the amount of air admitted, the compression effected, and the timing of the ignition.

To effect governing, two or more of these features are, as has been seen, generally changed simultaneously.

In the hit-or-miss system the gas alone, or the gas and air, are shut off entirely at excessive speeds, but other features remain unchanged.

In throttling an already completed mixture the gas and air volumes are changed proportionally, and, thus, the quality of the charge remains unchanged, but the compression will be diminished.

By having the gas and air throttle controlled separately the quality of the mixture may be changed, but the quantity unchanged, and, thus, the compression unchanged.

Between these proportions the quality of the charge may be changed to any extent, resulting in a more or less decreased compression. It may even be possible to dilute the charge to such an extent that its quantity and compression become greater at reduced loads.

In Fig. 112 are represented a full-load and a light-load card from a throttling engine.

The initial pressure, the compression, and the mean effective pressures are as follows:

Of the full-load card—

Initial pressure. $p' =$ 14 pounds per square inch.
Compression pressure. $p'_c =$ 104 pounds per square inch.
Mean effective pressure. $p'_m =$ 70 pounds per square inch.

Of the light-load card—

Initial pressure. $p'' =$ 10 pounds per square inch.
Compression pressure. $p''_c =$ 72 pounds per square inch.
Mean effective pressure. $p''_m =$ 20 pounds per square inch.

Assuming V_f and V_l to be the volumes of the mixture admitted per stroke, of standard temperature and pressure, respectively, at full and at light load; and assuming the temperature of the charge to be at both occasions the same, then we have

$$\frac{V_f}{V_l} = \frac{p'}{p''} = \frac{14}{10} = 1.4.$$

This ratio represents also the ratio between the heating-value admitted per stroke at the two instances. The ratio between the work performed during the full load and the light load period is represented by the ratio between the mean effective pressures, and is

$$= \frac{p'_m}{p''_m} = \frac{70}{20} = 3.5.$$

Hence, the efficiency is $\frac{1}{1.4} \times 3.5 = 2\frac{1}{2}$ times as great at full load as at light load.

The main cause of the poor efficiency at light loads is the slow combustion of the charge, due to the low compression. It would, therefore, be desirable to increase the compression at light loads, which cannot be done, however, excepting by diluting the charge with more air; and the question arises, if a quicker combustion and higher efficiency will be obtained from a poorer mixture and higher compression. With respect to some fuel-gases this appears to be the case, and, on this account, a so-called constant-quantity regulation is in many cases the more economical.

Fig. 113 represents a full-load and a light-load card from a constant-quantity regulation, and, although the combustion at

light load is improved upon compared with the condition in Fig. 112, the maximum pressure occurs, even here, too late in the stroke. The suggestion is then near at hand, to ignite the charge earlier

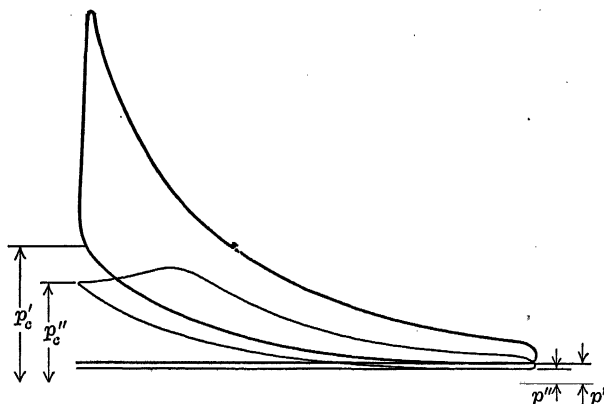



FIG. 112.—Constant-Quality Regulation.

at light load, so as to obtain a more complete combustion at the beginning of the pressure-stroke. Regulations of this order, combining a constant-quantity regulation and advancing ignition, have been tried, and the appearance of the cards would indicate

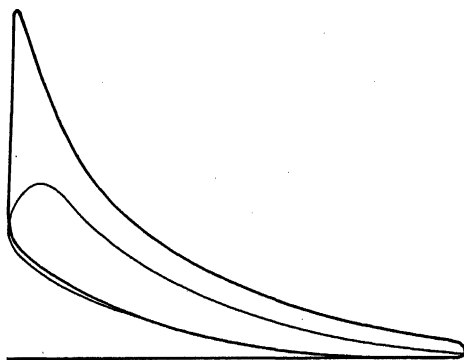



FIG. 113.—Constant-Quantity Regulation.

an improvement upon the simple throttling, or cut-off, governing. Any decided improvement in economy has, however, failed to appear in practice.

The diagram Fig. 114 represents a full- and a light-load card of a constant-quantity regulation, with advancing ignition on light loads. It will be noticed how in the light-load card, represented in fine lines, the ignition takes place early in the stroke, causing the initial pressure to become quite high. The question may, however, be asked, if the increased frictional work due to a high resistance on the compression stroke does not, actually, offset what little advantage there is derived from a more perfect combustion.

It was brought out, in connection with the subject of the balancing of the reciprocating parts, that in heavy engines, particularly in engines of the tandem type, a high compression is of

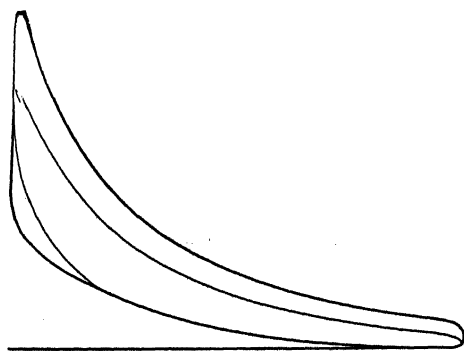



FIG. 114.—Constant-Quantity Regulation with Advancing Ignition.

advantage for quiet running of the engine. It would therefore seem that the constant-quantity regulations, in which the compression remains the same for all loads, would particularly be desirable for heavy engines.

It being a fact that a weak charge requires a higher compression for rapid combustion than a richer one, it has been suggested and tried, to compress, at light loads, a charge of a highly diluted mixture to a higher pressure than the regular charge at heavy loads. In other words, the rich mixture at heavy loads is throttled to the required extent for avoiding pre-ignitions, and the weaker mixture is admitted at a greater density to obtain

a high compression. In the diagram Fig. 115 are shown a full- and a light-load card representing a regulation of this kind,* in combination with an advancing-spark regulation.

Theoretically, the combination regulating systems appear to be ideal, but, practically, the advantage gained in a higher efficiency is often offset by the greater complications they involve. There are, however, cases when the nature of the fuel, and the load-conditions, are such that an improved combustion is of importance, disregarding whatever improvement in efficiency may be effected.

In throttling engines drawing the fuel-mixture from a mixing-chamber, particularly when the latter serves two or more cylinders, there is often, at light loads, difficulty from back-firing into the

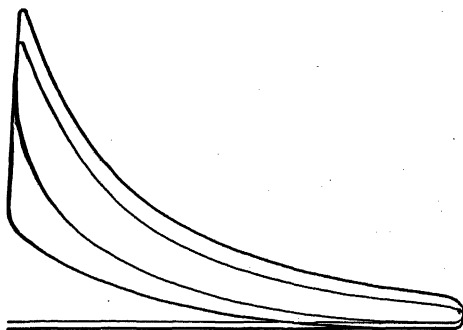



FIG. 115.—Regulation with Increasing Quantity at Light Loads.

mixture. It is due to slow combustion of the fuel when insufficiently compressed; the charge still holding fire when the inlet valve opens for a following suction-stroke. With fuels not readily vaporized, or poorly mixed, difficulties of this nature will be serious. In such cases a high compression and an early ignition will be very essential, and the combination governing system may be used to good advantage.

Factors that Enter the Problem of the Governing of Gas-

* This regulation was a feature of the Letombe gas-engine built by the Société Anonyme d'exploitation brevets Letombe at Lille, and exhibited at the Paris Exposition.

Engines.—The conditions under which the gas-engine operates make the perfect governing of the engine much harder to accomplish than is the case with steam-engines. The disturbances that militate against a good regulation of the speed, disregarding those due to explosion waves about which not enough is known to tell exactly what influence they may have on the regulation, are, principally, pre-ignitions, back-firing, and fluctuations in the explosion-pressure.

Pre-ignitions and back-firing, which may be considered as being of only temporary nature, are readily guarded against when the cause of the disturbances is understood; and after an engine is once adjusted to the conditions under which it is working, such disturbances are, generally, practically eliminated.

Pre-ignitions, or self-ignitions, are often due to one or other of the following causes:

Excessive compression of the fuel used; an uneven gas-mixture, a too readily ignitible rich charge being occasionally supplied; an over-heated exhaust-valve, piston, ignitor, or some protruding and poorly jacketed part of the combustion-chamber; a poor grade of cylinder oil allowed to leak past the piston and becoming carbonized in the combustion-chamber; foreign matter and impurities admitted with the gas or air charge, collecting in the combustion-chamber and holding the fire; a too rich fuel-charge which, due to incomplete combustion, leaves a carbon deposit in the cylinder.

If due to the over-heating of some part in the combustion-chamber, a more thorough water cooling, or the removal of some unnecessarily protruding part may become necessary to stop pre-ignitions; or, occasionally, the reduction of the engine-speed may help matters.

The usual causes of back-firing into the mixing-chamber are, in general: slow combustion due to a low compression, a fuel poorly vaporized, or leaky valves. The effect from back-firing may be very much reduced, or entirely eliminated, by doing away with the mixing-chamber, and effect the mixing of the fuel and air first at the inlet-valve. Should the inlet-valve leak, it would, of course, with this arrangement still be possible that the flame

might throw back into the gas and air pipes, but any serious disturbance will not be caused.

In Fig. 116 is shown a continuous diagram of the explosion-pressures obtained for a series of explosions. Diagrams of this description are readily obtained by advancing the paper-drum of the indicator, steadily, for a series of explosions, while the pencil-arm records the height to which it is driven by the explosion-pressures. The diagram illustrated, which was taken during a

FIG. 116.—Continuous-Explosion Diagram.

period of partially light and partially heavy load, but, at each period of perfectly steady load, shows the fluctuations in the maximum pressure from cycle to cycle.

The causes of the variation in the explosion-pressure, under constant load-conditions, are of a variety of kind, and cannot readily be guarded against.

The more evident causes of the fluctuating pressures are:

An uneven mixture, due to pulsations in the gas, the atmosphere, or the exhaust; an uneven compression, due to an unsteady governor, due to pulsations in the mixture supply, or due to poorly guided valves, causing the valves to seat themselves more or less tight; variations in the point of the ignition, probably due to the springing of, or due to the inertia of, the ignition-device, causing the spark to be timed unevenly.

With a reasonably heavy fly-wheel, the effect of the fluctuations

in explosion-pressure is not generally serious enough to impair the steadiness of the speed of the engine, but, in order to obtain the most effective governing, the effort should be to eliminate as many of the enumerated disturbing influences as possible, without the introduction of complications of detail, that, on the other hand, will in themselves become objectionable.

CHAPTER XIII

ENGINE AUXILIARIES

Carbureters.—When liquid fuels are used, a fuel-vaporizer, or what is commonly called a carbureter, becomes a necessary auxiliary to the gas-engine. The three most generally employed modern types of carbureters for gasoline are shown in Figs. 117, 118, and 119. The one illustrated in Fig. 117 is of the type in which the gasoline is supplied to the vaporizer under a slight head; a check-valve being utilized to check the flow during the intervals between the suction-strokes. The check-valve is, in the apparatus illustrated, made large, serving also to check the return flow of the carbureted air. This type is often used in connection with two-cycle engines, in which the carbureted air is under pressure during the forward stroke of the piston. The gasoline-connection is made at *I*, and the supply is adjusted, as to quantity, by means of the needle-valve, *N*, which may be set by trial to give a suitable opening for the admission of the fuel.

Carbureters of this type are simple, but they are liable, at varying loads and when the gasoline-pipe is of some length, to give trouble. There being no storage of gasoline near the spray opening, pulsations in the gasoline supply-pipe affect the tillflow of the fuel differently at different periods. This difficulty may be overcome by fitting in the fuel-pipe near the carbureter a constant-level reservoir from which the fuel will be drawn directly into the supply-opening to the carbureter.

It must be borne in mind that, in order that the gasoline-engine shall operate properly, the fuel required for the proper mixture at any load-condition must be supplied uniformly, and that the engine will never act well if there is the slightest cause for sudden and extreme changes in the composition of the mixture, which are not called for by any variation in the load. The more sensitive the governor is, other conditions being equal, the more sensitive the engine will be for pulsations in the fuel-supply.

Assuming that, due to disturbances in the fuel-supply, a heavy charge be admitted at a period of light load, it is evident then that a following charge would, through the action of a sensitive throttling governor, become extremely weak, and apt to cause back-fire into the mixing-chamber due to slow combustion. Again, alternate very rich charges are apt to cause pre-ignitions due to an over-supply of the fuel.

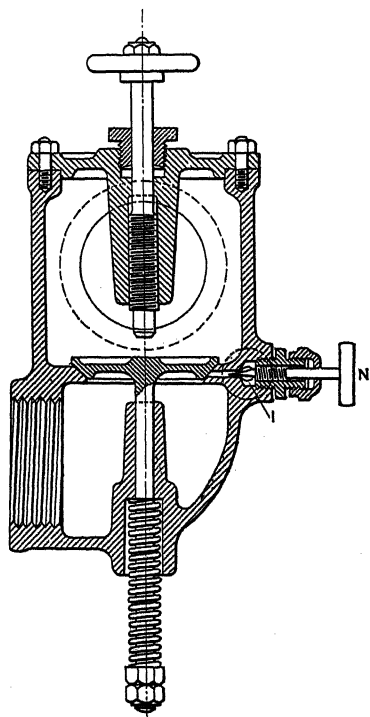



FIG. 117.

Fig. 118 is a constant-level reservoir-carbureter. In connection with this carbureter there is a small gasoline pump driven by the engine, which supplies a suitable amount of the fuel from the fuel-tank into the reservoir *R*; keeping it constantly filled to a certain level, one-half to one inch below the spray-nozzle *N*. Whatever amount of fuel the pump supplies above what is used by the engine will be returned through the overflow pipe *O* to the fuel-tank. At each suction-stroke of the engine, gasoline will be drawn into the vaporizer-chamber, due to the slight vacuum created there, and it will be absorbed by the air entering through the air-supply pipe *A*. Before starting the engine, the fuel must, of course, be pumped by hand into the reservoir, to supply the requirements for the first explosions.

This carbureter is often used to advantage in connection with stationary four-cycle engines.

Fig. 119 is a type of float-feed carbureter, which is most generally employed for all classes of gasoline engines. *C* is a small float-chamber, in which the fuel is held at a constant level

by means of the float, *F*, and inlet valve, *V*, operated by it. From the float-chamber the fuel is supplied to the nozzle, *N*, through which it is sprayed during the suction-strokes into, and absorbed by, the passing air-current entering through the air-supply port *A*.

It is often the case that the carbureted air leaving the carbu-

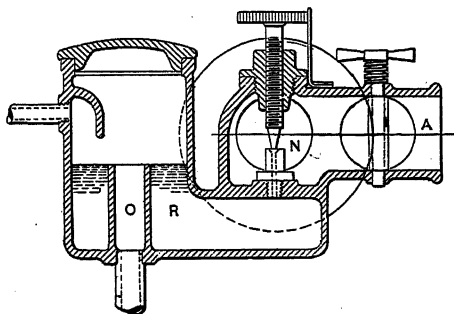



FIG. 118.

reter is too rich in gasoline-vapor for complete combustion, and it becomes necessary to mix it with additional air. The secondary air-supply is, in the apparatus illustrated, taken in through the orifice at *O*, and it may be throttled to any extent required, by

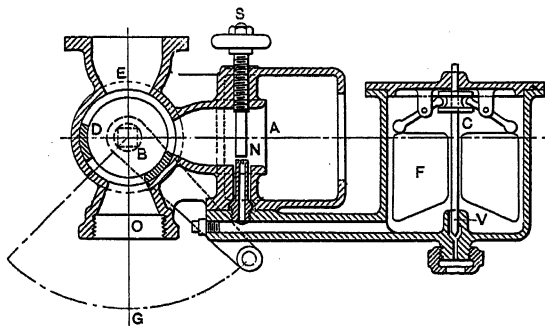



FIG. 119.

bringing the throttle-valve handle more and more over toward the position *B G*. After this position is reached the cut-off valve-bridge, *D*, comes into action, closing the inlet, *E*, to the cylinder.

There is no needle-valve used for regulating the orifice of the

spray-nozzle; the latter being made permanently of a size to correspond to the requirements of the engine. The screw-spindle *S* allows, however, to some extent, an adjustment to be made.

The by-passing of air through the secondary air-inlet is necessary in engines of a considerably varying speed, in order to overcome the tendency of the suction spray-nozzle to supply an excess of fuel at high piston-speed, when the vacuum in the mixing chamber becomes high.

The carburetion of the air will be more effective when dry and slightly heated air is used for absorbing the fuel-vapor; and the air-supply is, therefore, often drawn from a place where the heat from the engine-cylinder or exhaust-pipe will have for effect to pre-heat it to some extent. The pre-heating of the primary air-supply will, particularly, be desirable when, as often is the case, only part of the final air-charge, one-third or less, is expected to vaporize the full quota of gasoline.

Alcohol Carbureters.—The same carbureters which are used for gasoline may, with certain restrictions, also be used for vaporizing alcohol-fuels. For alcohol the pre-heating of the air is much more necessary than with respect to gasoline-fuel, because the former fuel vaporizes much more slowly. The object should also be to minimize the vacuum in the vaporization-chamber, because a high vacuum tends to chill the air and thus prevent it from readily absorbing the fuel. On this account, the port-openings through the apparatus, for both air and fuel, are required to be more ample than in the case of some more readily vaporized fuels. The entire air-supply should pass over the spray-nozzle, or, at most, only an unimportant part of the supply be by-passed.

Principal Auxiliaries of the Automobile Motor.—In Fig. 120 is represented a modern automobile engine, in connection with its fuel-supply system and cooling system. The former consists of the gasoline tank, from which the fuel is supplied to the float-case and from there to the carbureter; always at a constant head. In the carbureter a fuel throttle is applied, by means of which the speed and power of the engine can be regulated by the driver;

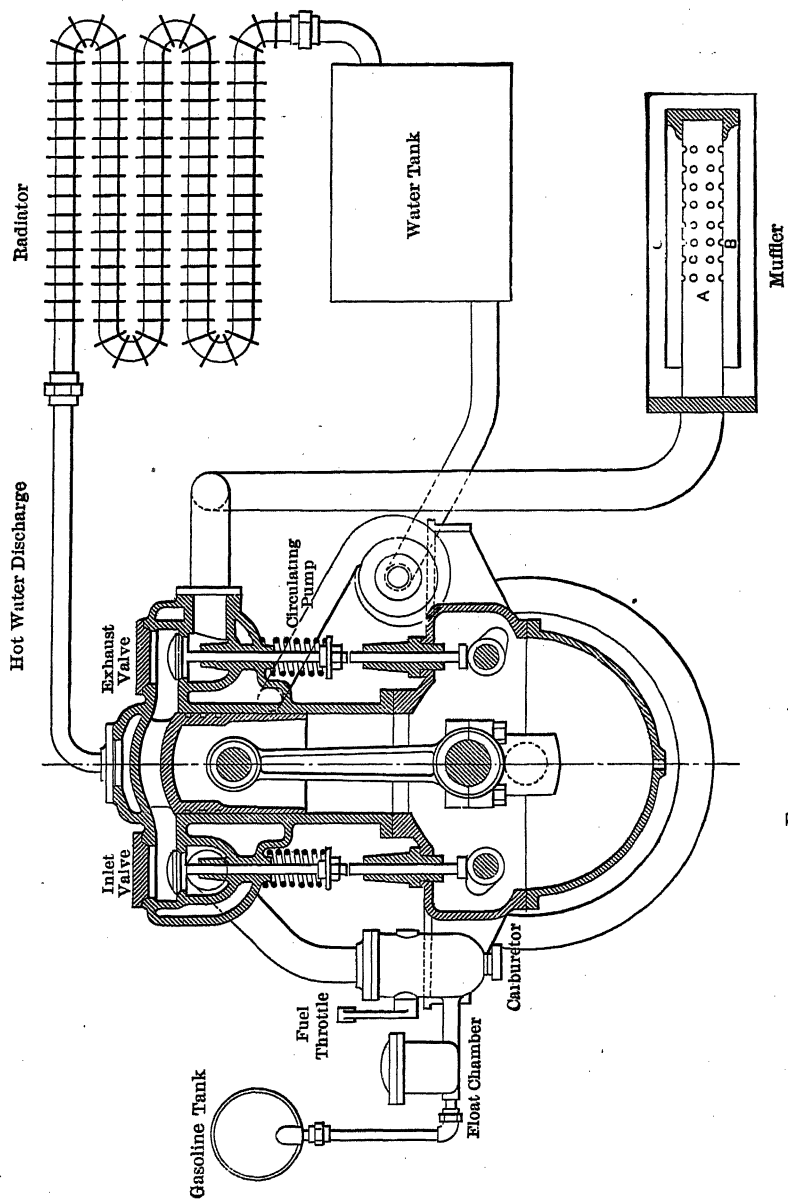



FIG. 120.—Automobile Engine and Auxiliaries.

or the fuel may be cut off to the extent of merely keeping the motor, empty, in motion. The gasoline tank is, in practice, not always located so as to feed by gravity, as in the figure, but a slight air-pressure may, by means of a hand-pump, be created in the tank, by which the fuel is forced out, and in to the float case, with equal certainty wherever the tank may be located.

High-power machines must always be cooled by means of a water-cooling system, since the cooling-surface required for the dissipation, directly to the air, of the heat that necessarily must be carried off is, in these, far in excess of that which the engine itself can well present. A radiator composed of a number of ribbed pipes, can however, always be built to present any surface required; and when in motion, the air then being set in a rapid circulation through its coils, such an apparatus is very effective for dissipating heat.

A small rotary pump is used to effect a good circulation in the cooling-system. The hot water passing off from the highest point of the jacket is carried through the radiator, in which its temperature is rapidly lowered and after having been cooled it collects at the bottom of the water-tank below, from which it is again drawn by the pump and forced in at the lower part of the engine jacket.

The Exhaust Muffler.—An exhaust muffler, or silencer, is shown connected to the discharge-pipe from the engine in Fig. 120.

In order to prevent the noise incidental to the escape into the atmosphere of the exhaust from the gas-engine, the remaining heat-energy of the gas, its capacity for doing work, which will be transferred into noise at its escape, must be dissipated.

Its pressure might be reduced, or its volume increased, by which two alternatives its sensible heat becomes reduced. Further its potential energy must be dissipated by reducing the velocity of the escaping gas.

These requirements are effected by changing, gradually, through expansion, the heat-energy of the discharge into velocity of the gases, and by reducing this velocity, by means of baffles, to a desirable limit, before the gases are allowed to escape. Of

course a direct cooling of the gas in the exhaust pipe, if practical, will greatly help to reduce its heat-energy and to silence the exhaust.

In the apparatus illustrated, the gradual expansion of the gas is effected by allowing it to escape through numerous holes shown in the pipe *A*. The velocity acquired by the various streams of gas when expanding in these nozzles is then readily baffled in the chamber *B*, and still further in the return-chamber *C*.

The muffler, unless it is made quite large, always throws some back-pressure on the engine, and it is on this account not a very desirable apparatus, but it is, in many cases, a necessary auxiliary to the gas-engine.

Ignition Devices.—The modern method for igniting the charge in the gas-engine cylinder, excepting in certain classes of kerosene- or oil-engines which are self-igniting, is by means of an electric spark. The requisite current may be obtained in various ways. By means of a common cell-battery, by a storage-battery, from an electric service-circuit, or by means of a small special dynamo or induction-magneto.

The electric spark is formed in two ways. By the so-called jump-spark system, and by the make-and-break system.

In the former system there is provided, in the secondary circuit from an induction-coil, a gap between two sparking-points in the cylinder. When thus the primary circuit, in series with the battery, is opened or closed a spark will be formed between the sparking-points—it, so to speak, jumping across the gap between the points. Hence the name of the system.

Fig. 121 illustrates the general scheme for the jump-spark system. *I* is the induction-coil, *B* is the battery, *V* the wiper opening or closing the primary circuit by coming in contact or out of contact with the contact-piece *C*. *S* represents a standard spark-plug screwed into the head of the cylinder and connected at its terminals with the leads from the secondary winding of the induction-coil.

With the wiper *V* touching the contact *C*, the primary circuit may be opened and closed a series of times by means of an ordinary vibrator, such as is used on a *Rhumcorf* coil, in which

case a number of sparks are formed at the sparking-points. Ordinarily, however, the vibrator may be left out and then two sparks only will be obtained, one when the wiper V closes and one when it opens the primary circuit.

The Make-and-Break System.—In the jump-spark system the current must be of high tension to cause a spark to form between the sparking-points; these being generally set $\frac{1}{32}$ to $\frac{1}{16}$ of one inch apart. In the make-and-break system the sparking-points are in contact before the time for a spark. When the circuit then is

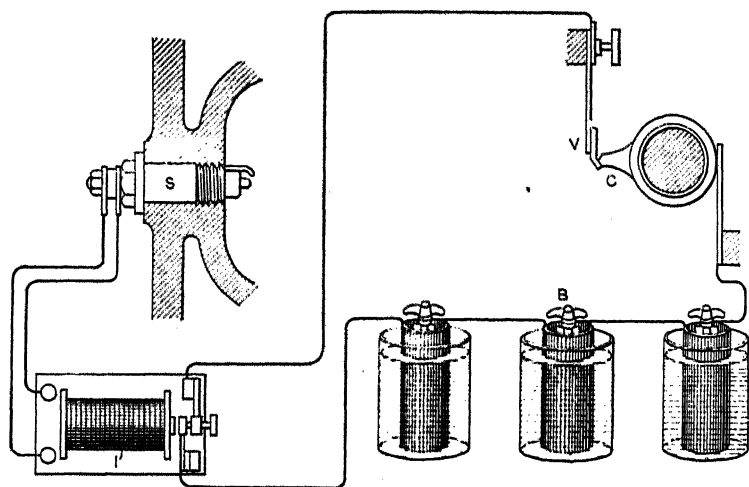



FIG. 121.

broken, suddenly, by moving apart the contact-points, a good spark will be formed with a current of less tension than that required in the jump-spark system, and the deterioration of the sparking-points will be less. The primary wire of an induction-coil is often inserted in the circuit, as in Fig. 122, in order to intensify the spark.

Fig. 122 represents the arrangement of an ignition device of the make-and-break system. In the circuit from the battery, B, is inserted the induction coil I and the leads, $l\ l_1$, connecting with the fixed and movable electrode. Ordinarily, the circuit is open by having the movable sparking-point thrown back in the position

a by means of the coil-spring *S*. *C* is a crank secured to the end of the valve-gear shaft, and it carries one end of the pick-blade lever, *L*, whose other end is secured by the pin *d* to the link, *H*. By the same pin is also secured the pick-blade, *P*, which is also held in position by a stiff spring *T*. When the shaft revolves in the direction of the arrow, the pick-blade will, at the proper time, just before firing, engage with the lever *M* and swing it toward the right, thereby closing the electric circuit; the spring *T* serving to press the contact-points firmly together. At the time for firing

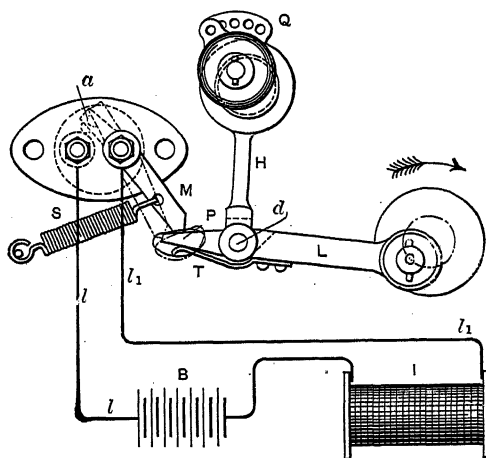



FIG. 122.

the lever *M* will disengage and suddenly break the contact between the sparking-points, causing a spark to be formed.

Instead of the battery, a small generator driven by the engine may, of course, be used, and in such a case the battery, *B*, and inductive coil, *I*, would be switched in the circuit only for starting the engine.

The inertia of the make-and-break mechanism tending, at high speeds, to make the timing of the spark more or less uncertain, it is apparent that the jump-spark system will be the more suitable for engines of a high number of revolutions.

Magneto Ignition.—The latest and simplest device for generating a current for ignition is by means of the induction-magneto.

The *Bosch* type of magneto, which is now often found to work in connection with all classes of engines, consists simply of a small Siemens armature which is made to oscillate between the poles of a permanent steel-magnet. Fig. 123 shows the complete arrangement of the ignition device in connection with this apparatus. *M* is the magneto. On the armature-shaft is secured a *T*-crank which is pulled positively to its neutral position by the springs *S S*. *C* is the cam-shaft of the engine. On the end of it there is secured the crank *F* carrying one end of the pick-blade lever *L*. When the shaft revolves in the direction of the arrow, the pick-

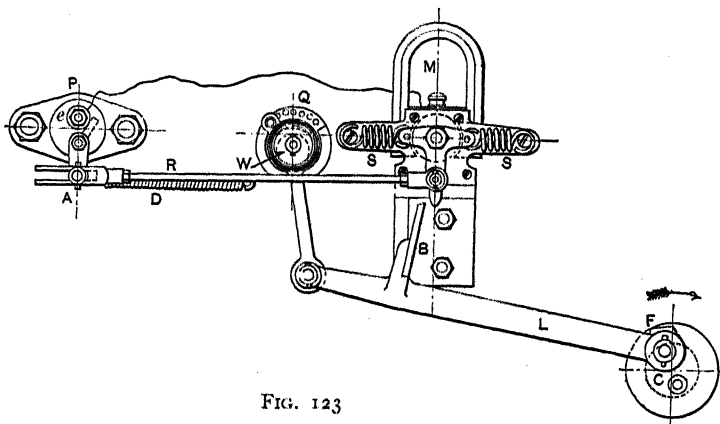



Fig. 123

blade *B* will, at its motion toward the right, engage with the *T*-crank, carrying it out of plumb toward the right. At the proper moment for firing, the *T*-crank is disengaged and is pulled quickly back by the springs *S*. A current is thereby generated in the windings of the armature, which is led to the stationary electrode *e* of the spark-plug *P*. At the moment when the *T*-crank is disengaged and swings to its normal position, the forked end of the push-rod *R* will give to the crank-arm *A* of the movable electrode a blow which causes it to swing to the left, thereby momentarily breaking the contact between the sparking-points in the cylinder. These points are normally held in contact by means of a weak coil-spring *D*, which is in the figure partly covered by the push-rod. In returning to the left the pick-blade

clears the end of the T-crank, and by lowering or raising the left-hand end of the pick-blade lever, by means of the link, eccentric and hand-wheel W, the release of the T-crank can be made to occur earlier or later, according to the desire to advance or retard the spark. The timing of the spark is thus accomplished by turning the hand-wheel W to the right or left and locking it in the various holes of the quadrant Q. The timing-device of the igniter, Fig. 122, is very much of the same arrangement.

In some makes of engines the timing of the spark is accomplished automatically by the governor; the object being to effect

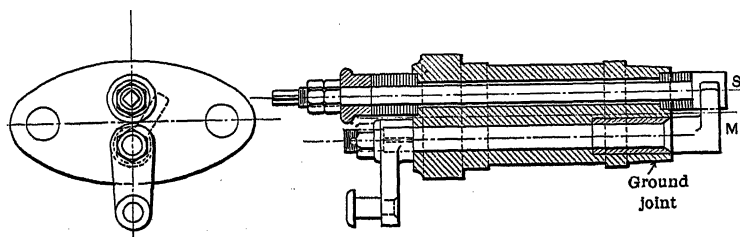



FIG. 124.

an earlier ignition at light loads, when the combustion is slow, due to a lean mixture and low compression.

In Fig. 124 is shown the general construction of a spark-plug for the make-and-break system. The stationary electrode S must be carefully insulated, electrically, by means of mica or lava washers and bushings, whereas the movable electrode M stands in electrical contact with the housing. There is, therefore, only one lead required from the magneto to the stationary electrode as shown in Fig. 123.

Similarly, the lead $l_1 l_1$, in Fig. 122, may, if more convenient, be connected to any point of the engine which stands in electrical contact with the housing of the spark-plug of a construction as described.

CHAPTER XIV

VARIOUS ENGINE-TYPES

The Two-Cycle Engine.—The two-stroke cycle or the two-cycle engine is suitable, chiefly, when the main object is to obtain the greatest simplicity in a small engine, or the greatest amount of power by the smallest weight in a large engine.

Figs. 125 and 126 show the main features of the two-cycle engine in its simplest form, and its action is as follows: During the up-stroke of the piston, as in Fig. 125, the charge of suitable mixture is drawn in to the closed crank-case *C* through the opening *A*, and it becomes compressed, slightly, during the down-stroke—to about 6 to 8 pounds above the atmosphere. When the piston, passing down, comes to a proper position near the end of the stroke, the exhaust port *E* opens first for the exhaust gases to pass out from the pressure side of the cylinder, and, later, the inlet port *I* will begin to open for new charge. Both ports become fully open when the piston arrives at the lower centre, as shown in Fig. 126. The charge will enter from the crank-case in to the cylinder, expelling the neutrals before it, until, soon after the beginning of the upward stroke, the inlet port first and then the exhaust port becomes closed by being covered by the piston, and the compression of the charge commences. When fully compressed, at the time the piston arrives at the top position, the charge will be ignited, and the power stroke will follow during the down-stroke.

Thus, every down-stroke will be a power-stroke and every up-stroke a suction-stroke. Two full strokes of the piston constitute one cycle.

A closer regulation of the charge and a more reliable action can be obtained by using an inlet-valve with adjustable spring-tension. Figs. 127 and 128 illustrate a small marine engine with such an inlet-valve, of the design of Smalley Bros. Co., Bay City, Mich. The speed-regulation of engines of this type is sometimes

effected by the throttling of the charge in the inlet-valve, in which case means are provided for preventing the inlet-valve from lifting to its full lift, when a reduced speed is required.

The compression of the charge, instead of being performed by the working piston, may, of course, be accomplished by a special

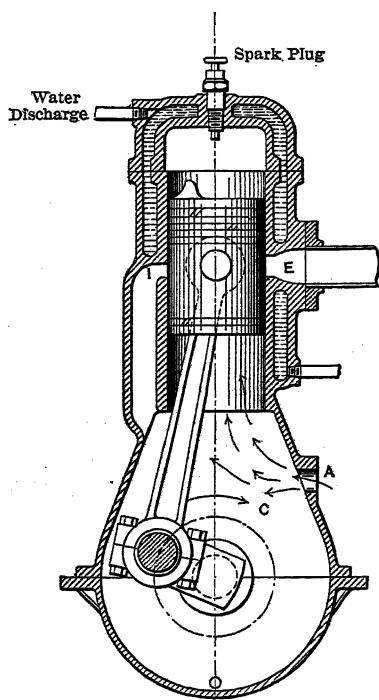
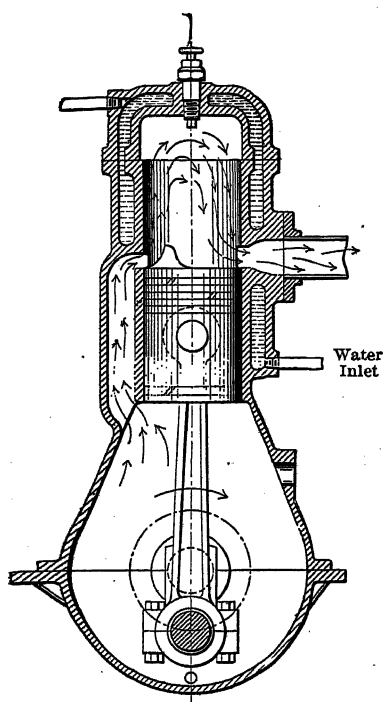



FIG. 125. FIG. 126.

compressor-cylinder, and in large two-cycle engines this will be nearer at hand.

The Koerting Two-Cycle Engine.—The Koerting double-acting, two-cycle engine is a type in which the air and gas for the charge are compressed and delivered separately by auxiliary pumps driven by the main engine shaft. The pump-pistons are so proportioned and their motion so timed with respect to the main working piston that the proper amount of air and gas will be delivered to the working cylinder at the proper time. It is

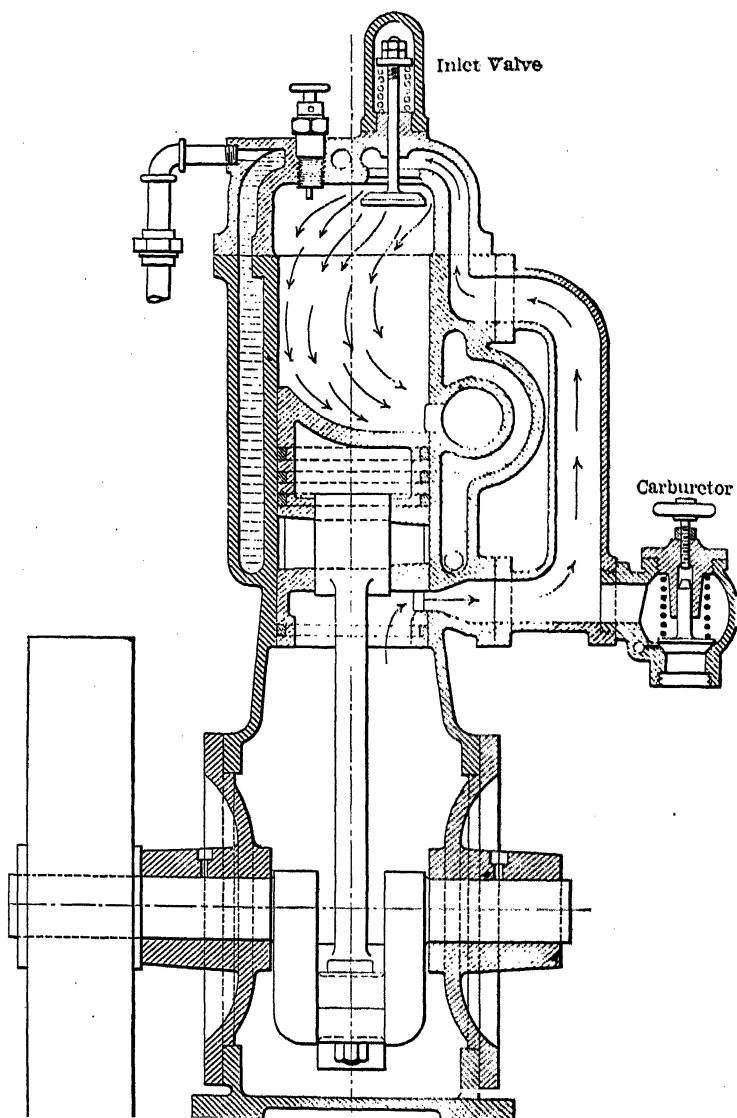



FIG. 127.

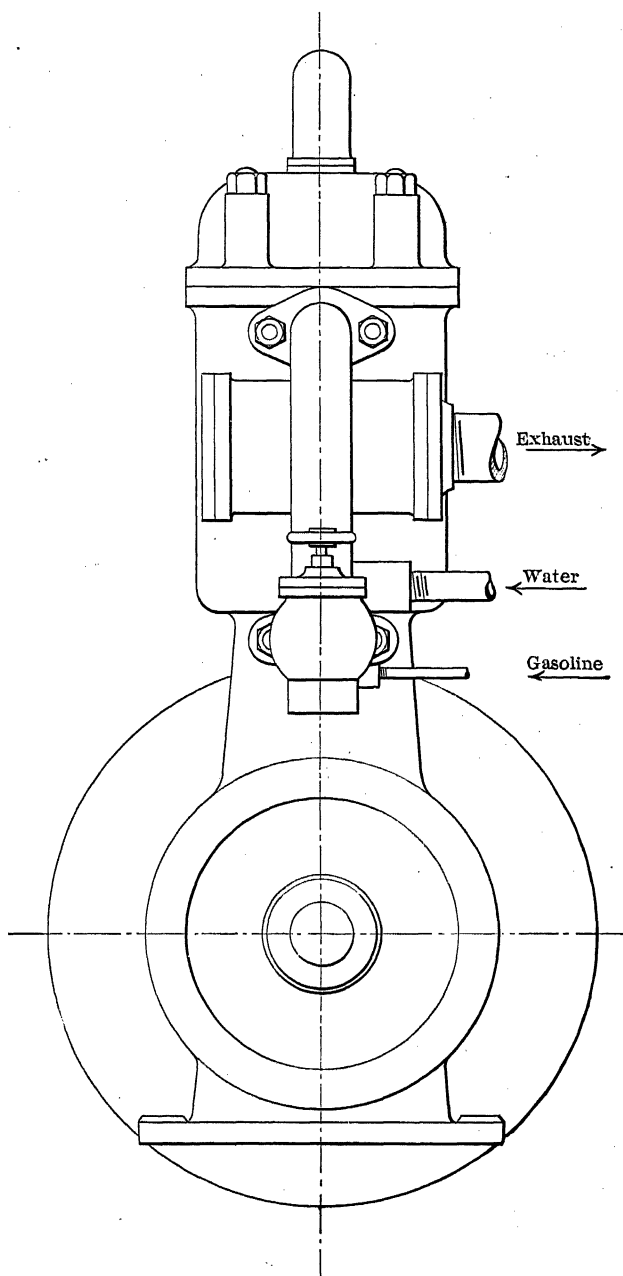



FIG. 128.

to be noted that the gases remaining in the cylinder from a previous expansion-stroke are not, in the two-cycle engine, removed by a following exhaust-stroke, and it becomes therefore necessary to resort to scavenging, in order to remove the neutrals which fill the cylinder after the expansion-stroke is completed. This is in the Koerting engine, as in two-cycle engines generally, accomplished by forcing a current of air through the cylinder while the exhaust port remains opened, and it becomes the function of the air pump to furnish also the necessary air for the purging of the cylinder.

Figs. 129*a* and 129*b* are a plan and a front view which show

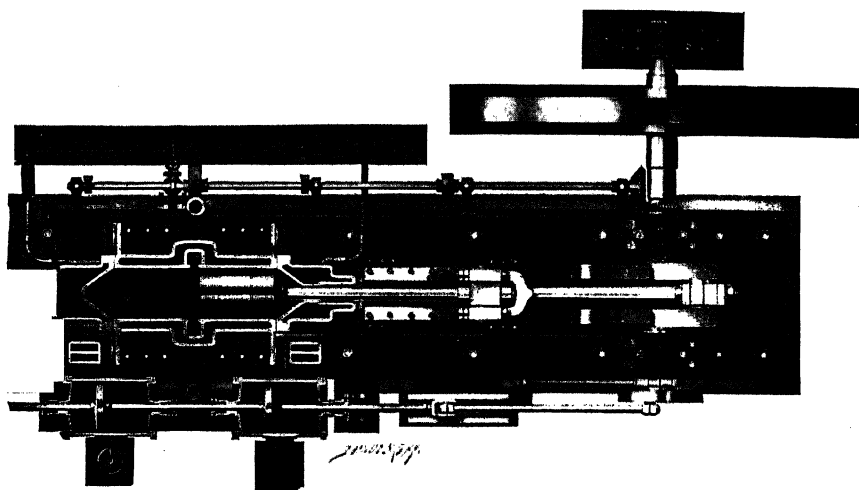



FIG. 129*a*.—Koerting Two-Cycle Engine. Sectional Plan.

the general arrangement of the Koerting engine, and in Fig. 130 the working cylinder and the pumps are shown diagrammatically.

As seen in Fig. 130, there is, for each end of the working cylinder, only one valve—the admission valve; the discharge being exhausted through ports, *M*, at the middle of the cylinder, and these are covered by the piston at all times, excepting at the end of each working stroke. The admission valve-casings *A* and *B*, are provided with separate air- and gas-ports terminating directly above the admission valves.

Referring to the compressor pumps: *C* is the air pump and *D* the gas pump, and their pistons are driven by a continuous rod from a crank on the main engine-shaft.

The action of both pumps is controlled by piston-valves—*E* the air valve and *F* the gas valve—driven by means of rockers and separate eccentrics from the main engine-shaft (an earlier type of valve gear which is shown in Fig. 129*a* used, however, one eccentric only for both valves). The admission to the air pump is at *G*, and the air discharge ports are at *H H*, while the admission to the gas pump is at *I* and the gas-discharge to the working cylinder at *K K*. By the paths of the arrows leading

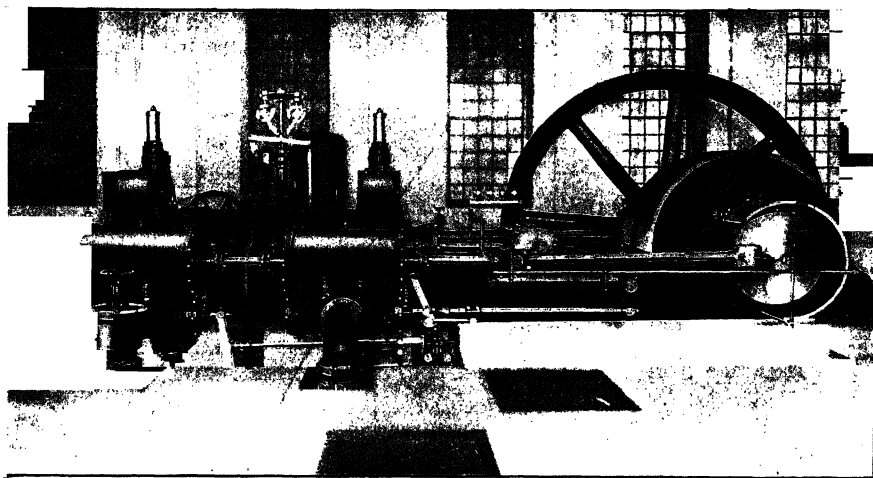



FIG. 129*b*.—Koerting Two-Cycle Engine. Front View.

from the discharge ports of the pumps are indicated the connecting channels between the pumps and the working cylinder, and it will be seen that the crank-ends of both air- and gas-pumps discharge through separate channels to the crank-end of the working cylinder, while the head-ends of both pumps discharge, also through separate channels, to the head-end of the working cylinder; *x x* are regulating valves to which reference will be made presently.

In Fig. 130 the working piston, *P*, is represented as having

arrived at the end of its forward stroke, and the exhaust ports are fully uncovered for the discharge to escape from behind the piston. When the piston occupies this position the pressure back of it, in the head-end of the cylinder, has become nearly fully relieved, and the head-end inlet valve, *B*, has opened a small amount. The crank driving the gas and air pumps being

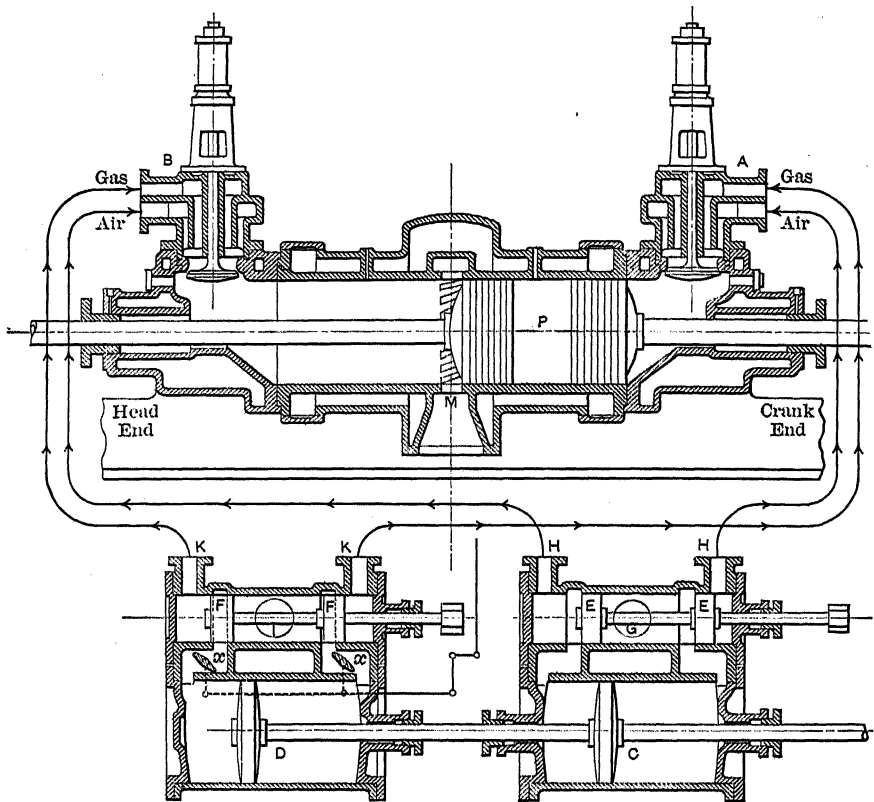



FIG. 130.—Koerting Two-Cycle Engine.

set ahead of the main crank about 110 degrees, the pump-pistons, when the main crank passes the forward centre, will occupy, approximately, the positions in which they are shown in the figure. The piston-valve for the air pump will be open for discharge at the head-end, and the pressure in front of the air piston

will be from 4 to 8 pounds gauge. When, therefore, the main inlet valve opens there will be an inrush of air from the air passage, which drives the burned gases, as completely as possible, out through the exhaust port and fills the cylinder with pure air. The piston-valve controlling the gas pump is, as yet, closed, but it will open immediately for supplying the fuel for the next charge

In the diagram, Fig. 131, the main crank and the pump crank are represented in positions corresponding to those of the working piston and pump pistons in Fig. 130. When the main crank passes the point *a* the exhaust valve will begin to open and it becomes fully open at the end of the stroke, at *c*. At *b* the main inlet valve begins to open, admitting the scavenging air-charge, as explained above, and at *d* the piston-valve for the discharge of the fuel from the gas pump to the engine-cylinder will uncover. This will occur, accordingly, a little later than the position in which the pistons are shown in Fig. 130. Both pump-pistons moving together, they will, from the point *d*, force air and fuel in to the working cylinder, in a proportion that depends on the relative size of the pistons and on the relative resistance the air and the gas will meet in the passages from the pumps to the working cylinder, and they will continue to force the charge in to the cylinder until the main exhaust ports close, at the point *e* of the diagram. From that point, until the main inlet valve closes, at *f*, there will occur compression of the charge between the pump-pistons moving one way and the working piston moving in the opposite direction. At the point *f*, which occurs approximately when one-third of the charging stroke has been swept through by the working piston, the pressure in the cylinder will be from 8 to 10 pounds, and from there, until the end of the charging stroke, the charge will be compressed by the working piston alone. When, at the end of the stroke, the charge is fully compressed ignition takes place.

The capacity of the air pump is determined to suit the volume of air required for scavenging and for charge, and this quantity remains the same for all load-conditions. The air-pump piston-valve is set so as to give the maximum efficiency at the compression;

admitting air to the pump cylinder soon after the beginning of the suction-stroke and opening for discharge as nearly as possible when the compression-pressure equals that in the delivery pipe. The gas valve, on the other hand, is set to deliver to the working cylinder a fuel-charge of the required pressure and at the correct moment, which is when the main crank passes the point *d*, Fig. 131, or when the pump-crank stands approximately 120 degrees

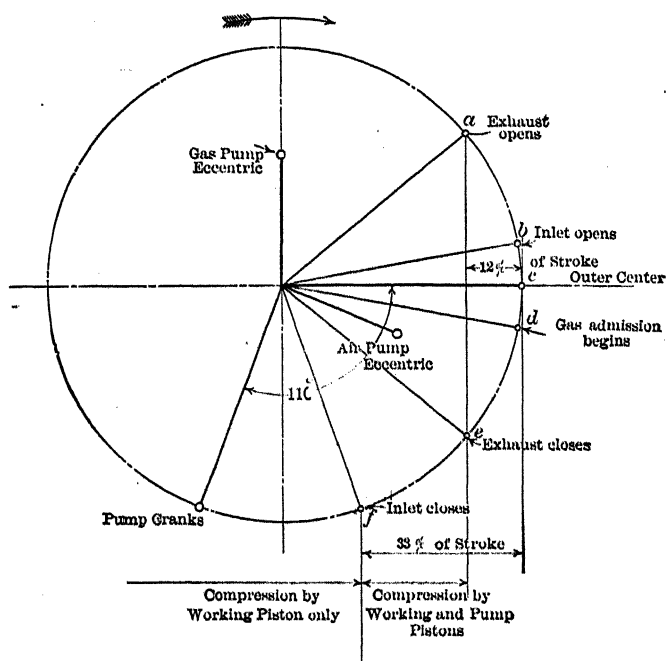



FIG. 131.

from the crank-end centre, *c*. The gas pump, therefore, will not begin to compress its charge at the beginning of the compression-stroke, but will return through the suction-port part of the charge which it has taken in, until the admission-port becomes closed at a point suitable for bringing the compression-pressure to that required at the point *d*.

Accordingly, with piston valves discharging at the outside edges as shown in Fig. 130, the adjustment of the eccentrics for

the air and gas pumps relatively to the crank will be, approximately, as shown by the diagram, Fig. 131.

The Koerting engine is governed on the principle of admitting a variable quantity of fuel in a constant quantity of charge, and this is carried out by supplying the fuel to the working cylinder at a pressure which varies in accordance with the load. The function the governor performs is to regulate, according to the momentary requirements, either the quantity of gas which is taken in to the gas pump for compression or the volume of compressed gas when it is being delivered to the working cylinder; either one or both.

The former may be done by means of a combination inlet and cut-off valve, or possibly by means of a shifting eccentric. The latter simply by throttling and by-passing the compressed gas when it is being delivered to the cylinder.

In the diagrammatical drawing of the Koerting engine, Fig. 130, the latter regulation is represented by the butterfly-valves *x x*, which are assumed to be under the control of the governor.

In the Koerting engine the piston travels 12 per cent of the stroke past the opening edge of the exhaust port, and during the time when the piston travels this distance, back and forth, until the closing of the exhaust port, which is just about one-fourth of the time for one revolution, the three functions, of discharging the waste gases, of scavenging, and of introducing the new charge, must be performed. An engine running 100 turns per minute would, thus, have only 0.15 of one second for performing these functions. It can readily be perceived, therefore, that the velocity with which the gases must be introduced in order to fill the cylinder in this short space of time must be considerable, and this circumstance is known to be the cause of a material loss through fluid friction, which may put a limit to the number of turns the engine can make, advantageously.

Fig. 132 is a diagram representing the exhausting and charging processes which take place at the end of each stroke of the piston, and it shows plainly the events occurring, from the time of the opening of the exhaust valve, until the compression of the charge by the working piston is in progress.

In order to promote the effective scavenging of the cylinder, the cylinder-head is formed as a deflector for the air current that enters through the admission valve on top of the cylinder. The object sought is to guide the general current of the charge in an axial direction so as to exclude, as much as possible, the intermingling of the new charge with the waste gases and to insure the latter's more effective expulsion.

Fig. 133 shows the general construction of the cylinder and

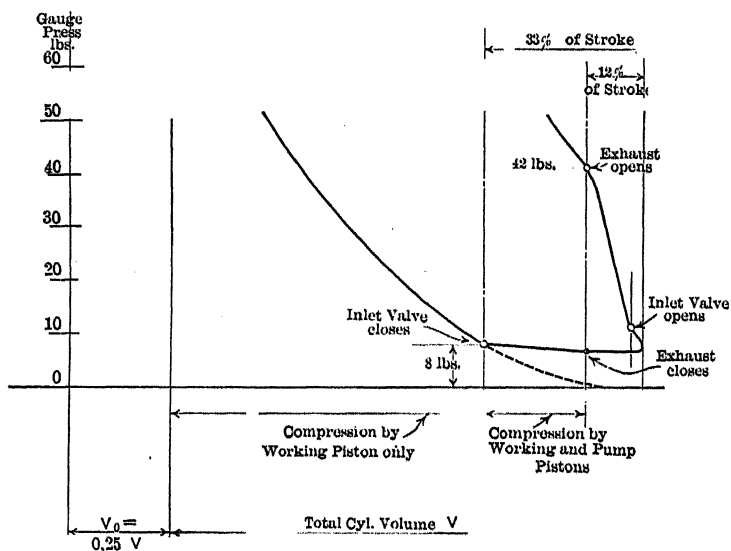



FIG. 132.

valve-gear of the Koerting engine built by the De La Vergne Machine Co. of New York.

The valve-gear shaft is driven, of course, in unison with the main shaft, by means of a pair of mitre-gears and the valves are actuated from this shaft by cams, and rockers which are fulcrumed on the valve-bonnets. The spiral gear for driving the governor will be seen in the illustration, at about the middle of the valve-gear shaft, but the governor itself is removed. The cooling system for the cylinder is plainly shown; the water being admitted at a low point at each end of the cylinder, and discharged from the very top of the middle of the same. It drains from there

through a pipe leading to a general cooling-water drain in the hollow base-casting, which discharges through the nozzle shown in the foreground in the illustration.

The piston is, of course, made hollow and is cooled by water admitted through a hollow piston-rod.

The engine is started by means of compressed air, which is admitted to the working cylinder by a small slide-valve, similarly as in an ordinary steam or compressed-air engine. The general

Fig. 133.

starting arrangement, which is practically the same for all types of double-acting engines, is described more in detail at page 404.

The ignition is controlled by means of an independent igniter shaft, the position of which, relatively to the valve-gear, can be advanced or retarded to suit the required time for firing.

The theory is often advanced, particularly with respect to the two-cycle engine, that a stratification of the different components of the charge in the cylinder will occur, so that after compression there will be found, nearest the piston, in succession, layers of

burned gases and of air, and from there the charge will successively grow richer and of a more perfect quality toward the inlet valve, near which the igniter generally is located. It seems, however, that the word "gradation" would more nearly express the occurrence; the likelihood being that the charge actually, under certain conditions, becomes more or less gradated in its composition—the inert elements and scavenging air being mixed in, in a greater quantity, toward the piston.

It is, of course, possible, by increasing the amount of air for scavenging, to increase the thoroughness with which the waste gases are expelled from the cylinder, though with increased loss due to negative work. It would, however, be at the risk of wasting part of the fuel through the exhaust port that the attempt would be made to fill the cylinder completely with fuel-charge. On this account the cylinder is generally not charged with fuel-mixture at a higher rate than to about 85 per cent of the volume of its working stroke. The complete charge in the two-cycle engine will, therefore, be approximately at par with that of the four-cycle non-scavenging engine.

The Oechelhaeuser Engine.—Another two-cycle engine, which, like the Koerting engine, is used extensively in Germany, where it is frequently installed for the operation of blast-engines, particularly, is the Oechelhaeuser engine, illustrated in Figs. 134*a* and 134*b*.

A feature of this engine is that its cylinder is equipped with two pistons; one reciprocating in the front end of a long cylinder, while the other works, adversely to the front one, in the rear end. The rear piston is, by means of a rear crosshead and yoke, side-rods, and a double set of main crossheads and connecting-rods, connected to two side-cranks of the same throw as the centre-crank to which the front piston is connected. This feature is plainly shown in the plan view, Fig. 134*a*.

The engine-shaft is, accordingly, equipped with three cranks of which the outside ones take, each, one-half of the force due to the impulse on the rear piston, and the centre-crank takes the full force due to the impulse on the front piston. For each revolution of the wheel there occurs, thus, two simultaneous

impulses; the one on the front piston moving toward the front and the one on the rear piston moving toward the rear. This is in effect, as far as the power, and the speed regulation during the cycle are concerned, the same as if there were for each revolution only one impulse, acting on the given crank arm, of twice the magnitude as that due one piston. Assuming that the reciprocating parts of an engine of this type were of the same weight as those of a four-cycle single-cylinder double-acting engine, then the speed regulation during the cycle would for both engines be the same, and the engines would require the same weight of fly-wheel. The balancing of the reciprocating parts of the engines would, however, be materially different.

It will readily be seen that the balancing of the Oechelhaeuser engine is nearly perfect, due to the fact that the two pistons are always moving in an opposite direction to each other. The only discrepancy from perfect balancing arises because the acceleration of the reciprocating parts moving toward the front is not, at all times, exactly the same as that of those moving toward the rear, due to the influence of the limited length of the connecting-rods. There may, of course, besides be some difference in the actual weights of the two sets of reciprocating parts.

On account of the practically perfect balancing, any tendency toward the rocking of the engine on its foundation is eliminated, but the tendency toward noisiness of its many main pins is still there; particularly if the reciprocating weights should be heavier than suitable for the compression-pressure and piston-speed employed. The piston-speed used in this engine is generally from 800 to 900 feet per minute.

It will be of interest to examine somewhat closely into the construction of the engine illustrated, as such an examination will bring out several features of advantage belonging to this engine-type.

The cylinder is constructed, as may be seen in the sectional view, Fig. 134*b*, of two plain inside cylinder-liners over which the two jacket-castings have been forced and connected by heavy flanges, at the middle of the completed cylinder. The inside flanged ends of the two liners butt together, and are held securely

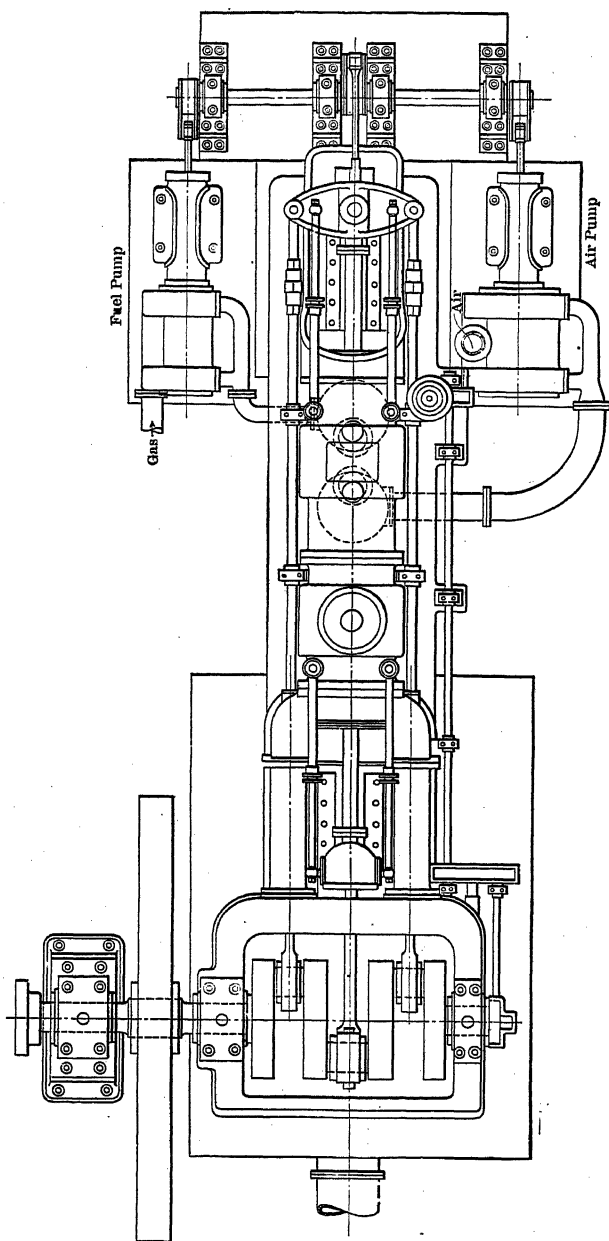



FIG. 134*a*.—Oechelhaeuser Engine. Plan View.

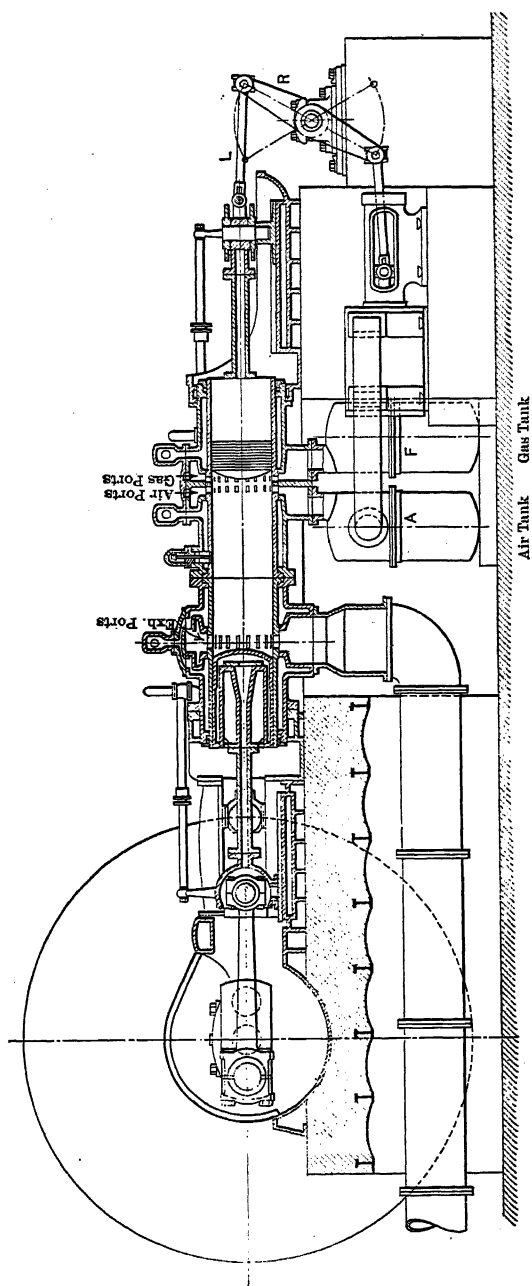



FIG. 134*b*.—Oechelhaeuser Engine. Sectional Elevation.

by the same joint that connects the jacket-castings, while their outside ends are free to expand and contract independently of the jacket, and, to allow this freedom of the liners, the water-joints at the outside ends of the cylinder are made by means of a packing, and by a gland which draws up against the ends of the outside castings.

The rear half of the jacket has, cast integral with it, the air and gas admission-channels, while the front half contains the discharge-channel for the waste gases. All these channels encircle and cut through the water-space of the jacket, so as to communicate all around the cylinder with rectangular ports which are cut in the liners, near the ends of the actual working space of the cylinder. At the bottom of the cylinder the channels connect, respectively, with the air supply, with the gas supply, and with the exhaust pipe.

At the middle of the cylinder there is, as seen, applied a check-valve, through which the compressed air for the starting of the engine is supplied.

The cylinder, thus completed, is centered at the front end, in the main engine frame and, at the rear end, in the rear guide-frame, and held axially by flanged joints. It will be evident that, as there will be no axial strain in the cylinder due to the working pressure on the pistons, there will never be any question as to the strength of any transverse section or circumferential joint between parts of the construction; and, as no strains are transmitted through the framework of the engine, there will be no difficulty in holding the rear part of the engine solidly to its foundation. These advantages are, most particularly, some of those belonging to this type of engines.

The pistons are water-cooled; the water-space being connected through hollow piston-rods, by means of telescoping tubes to the supply and discharge water-system.

In order to facilitate the removal of the pistons from the cylinder, there are inserted, between the crossheads and the pistons, flanged, short piston-rods that can readily be disconnected, and removed, in order to accommodate with the required space for sliding out the pistons.

The rear crossbar which connects the two side-rods is attached to the piston-rod by means of a pin-joint, which allows it to swivel the amount necessary to equalize the strains in the side-rods, and, thus, also the pressure on the side-cranks.

As in the Koerting engine, the cylinder of the Oechelhaeuser engine is scavenged with air, which in this engine is supplied from an air-tank, *A*, located directly underneath the cylinder. The fuel supply is also furnished from a similar tank, *F*. The air and the fuel supplies to these tanks are kept up by means of the air-

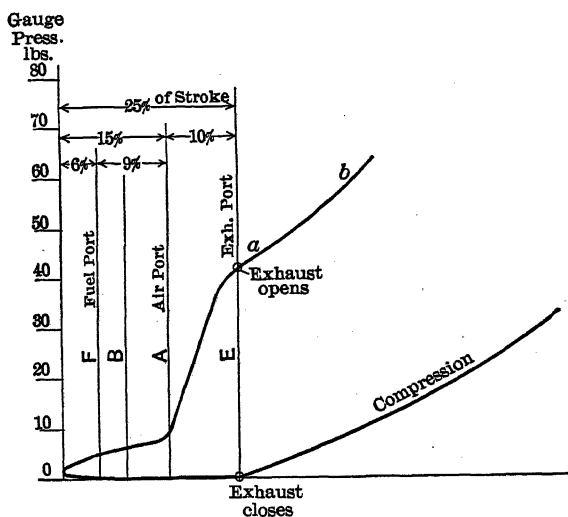



FIG. 134*c*.

and gas-pumps, which are both, generally, driven from the rear crosshead. In the engine illustrated they are driven by means of a rocking shaft, connecting to the rear crosshead by means of the rocker, *R*, and the link, *L*.

It will be evident how the pistons, by covering and uncovering the inlet and exhaust ports, will act as valves for the admission and release of the charge. The timing of the various events of exhaust, scavenging, fuel admission, and so forth, will most readily appear from the diagram, Fig. 134*c*. The lines *E*, *A* and *F* represent the cutting edges of the port-openings. The cutting

edge of the exhaust port, *E*, is approximately 25 per cent, the cutting edge of the air port, *A*, 15 per cent, and that of the fuel port, *F*, 6 per cent of the piston-travel from the end of the stroke. The ordinates of the diagram represent the approximate pressures existing at the various points of the crank-end of the pressure-card. The line *a–b* is the lower end of the expansion line. When the piston passes the line *E* the exhaust port becomes uncovered for the release of the old charge, which, after a period corresponding to about 10 per cent of the stroke, becomes of a pressure equal to, or below, that carried in the air-tank. At *A*, therefore, approximately 15 per cent from the end of the stroke, the ports communicating with the air-tank will uncover and the admission of the scavenging air commence. The admission of pure air for the displacing of the old charge is continued during a period of about 9 per cent of the stroke, until the fuel-port is uncovered, which occurs when the piston has arrived within 6 per cent to the end of its stroke. During the remaining part of the stroke, and until the fuel-port becomes covered during the return-stroke of the piston, air and gas, in a suitable proportion for a combustible mixture, are admitted. After the admission of the fuel-charge is completed, pure air only will again be admitted during a period corresponding to a piston travel of 6 per cent of the stroke, from *B* to *A*, until the air ports become covered. The exhaust has remained open from the time the piston, going out, at *E*, uncovered the ports, until they again become covered at the return of the piston to *E*. From there compression of the charge will continue to the end of the compression-stroke.

The main idea of this charging process, it will appear, is to obtain an effective mixture at the middle of the cylinder, where the ignition of the charge takes place, and to enclose this mixture at the ends of the cylinder, toward the pistons, by bodies of air.

It is evident, that should an overload be thrown on to the engine causing it to slow down, then, if the charging pressure were too high, the charge may have time to reach the exhaust ports, before these become closed by the front piston, and not only become wasted, but liable to cause premature ignition when ignited by the heat of the exhaust pipe. To preclude pre-ignitions

there is often applied in the exhaust-pipe, close to the cylinder, a jet of water that helps to keep the ports cool. In order to cool and effectively displace the hot neutrals, the air for scavenging is often carried of a quite high pressure (from 6, sometimes to 9 pounds). The higher the pressure the more effectively the air, at its expansion to the discharge pressure, will chill the waste products.

The relative proportions between the air and the gas admitted for the actual charge, at normal load, must, of course, be carefully adjusted to that suitable for the best mixture. To effect such a proportioning, even though the air in the tank, for scavenging purposes, is carried at a high pressure, there is often applied in the outlet from the tank a throttling valve, which, by means of an eccentric on the valve-gear shaft, is operated so as to throttle the air-pressure to that suitable for the mixture, as soon as the gas ports are ready to open. This arrangement is used by Borsig, who is a well-known builder of this type of engines.

The regulation of the Oechelhaeuser engine is arranged differently by different builders. The general idea is to effect, as nearly as possible, a constant-quantity mixture for all load-conditions. To accomplish this it would be required that the fuel-charge, only, be throttled, and by-passed to the suction side of the fuel-pump, more or less, to suit the load, but under such conditions the charge may, at light loads, become too lean even to ignite, and for that reason the air also must, generally, be throttled and by-passed by the governor.

The Borsig-Oechelhaeuser engine is, as a rule, fitted with two ring-valves surrounding the cylinder-liner, at the air- and at the gas-ports, and having port-openings registering with those in the cylinder. By means of these valves the air- and gas-ports may be adjusted, both by hand, to suit the full-load conditions, and by the governor, for the throttling of the charge at light loads. The air valve is arranged so as to throttle the air-ports at the top of the cylinder principally, by which arrangement the charge becomes richer near the igniters, and the firing more sure.

An engine of this type of a cylinder diameter 43.3 inches and 53.2 inches stroke, running 100 revolutions per minute on coke-

oven gas, is rated by the builders at 1,500 B.H.P., which is a very conservative rating. For an overload capacity of 20 per cent the engine would be of, approximately, 2,100 maximum I.H.P. The corresponding suction-displacement of the piston, per horse-power, is thus as much as 4.4 cubic feet per minute, or the mean effective pressure 54 pounds per square inch of the piston. The test of the Borsig-Oechelhaeuser engine included in Table XXXI records, however, an average mean effective pressure of 74 pounds. Good coke-oven gas, being of a heating-value practically the same as that of illuminating gas, should readily give a mean pressure even higher than this. However, as its heating-value generally fluctuates materially, a conservative figure should be used in estimates. The figure 74 has been recommended in Table XI for coke-oven gas.

The work required by the air and gas pumps is generally from 10 to 14 per cent of the total work indicated by the working cylinder, and this work, together with other resistances of the engine, must, of course, be deducted from the total indicated work obtained in the working cylinder for obtaining the brake horse-power of the engine.

The Indicated Power of the Two-Cycle Engine.—The question has been raised, in case a compressor-piston for supplying the compressed charge is driven by a two-cycle engine: should the total indicated power of the gas-engine cylinder be considered the indicated power of the engine; or should the power required by the compressor be deducted from the total power shown by the indicator-diagram, in order to obtain the indicated power of the engine?*

In the latter case, the indicated power would be at par with that obtained in a four-cycle engine-cylinder, which effects entirely its own compression. The matter resolves itself into a question as to whether the indicated power of the engine is that indicated by the gas-cylinder alone, or whether it is that indicated by the gas-cylinder and the compressor cylinder together; the latter being negative.

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* For the various opinions expressed on this question see *Zeitschrift des Vereins Deutscher Ingenieure*, for February, April, and May, 1905.

The value of the indicated power, or of the efficiency of an engine, is, of course, ascertained in order to serve as a basis for comparison with results obtained from other engines. In order that the results obtained from this special type of two-cycle engine shall be comparable with the condensing steam-engine, with the Diesel engine, or with some types of obsolete gas-engines, the indicated power must be that obtained by the gas-engine cylinder, without reduction for the work of compression, but in order that they shall be comparable with results from the four-cycle gas-engine the indicated power must be that indicated by the engine as a whole.

To compromise these requirements, it has been suggested by Mr. Diesel to express the power obtained in a two-cycle engine according to the following definitions:

The *total* indicated power (of the cylinder) = the full power represented by the indicator-diagram of the working cylinder.

The *net* indicated power (of the whole engine) = the indicated power of the working cylinder less the pump cards.

The brake horse-power = the work delivered to the end of the engine-shaft, and which can be taken off by means of a brake and brake-wheel.

The efficiencies would be:

The total mechanical efficiency = the brake horse-power divided by the total indicated power (of the cylinder).

The dynamic efficiency = the net indicated power (of the whole engine) divided by the total indicated power (of the cylinder).

The compressor-factor = the power represented by the pump cards divided by the total indicated power.

The engine-friction = 1 − the dynamic efficiency.

Four-Cycle Engines.—The Otto Engine.—The Otto engine is an old and well-known four-cycle engine operating on the hit-or-miss principle. The governor is of the fly-ball type, and it controls the admission or exclusion of the fuel by shifting the inlet cam-roller on or off the cam, according as the speed of the engine is below or above the normal.

The inlet- and exhaust-valves are arranged, one on each side of the combustion-chamber, in separate removable valve-casings,

and they are both operated by cams from a cam-shaft running along the side of the engine. This shaft is driven by means of a pair of spiral gears from the main shaft, at one-half the speed of the latter, and it carries besides the two valve-cams also, on
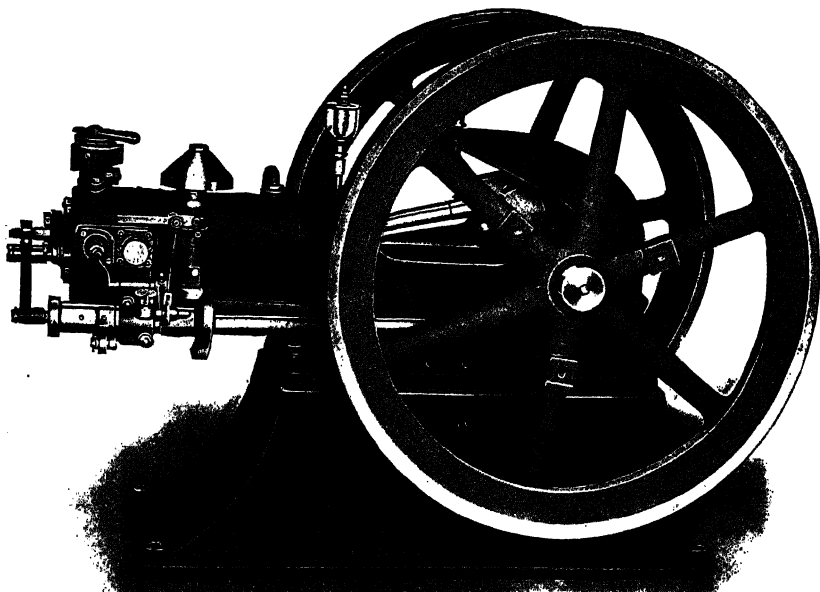
FIG. 135.

its extreme rear end, a small crank for operating the make-and-break ignition.

A fair idea of the general construction of the engine can be obtained from the illustration, Fig. 135.

The governor is shown in detail in Fig. 136, and the inlet valve-rocker carrying the cam-roller will be seen at the lower part of the figure. The position of the cam-roller is controlled by the governor by means of the bell-crank, in such a manner, that when the governor-sleeve is down then the bell-crank brings the cam-roller in a position to meet and engage with the inlet cam,

and thus effect the lifting of the valve. When, on the contrary, the sleeve, at excessive speed of the engine, rises to a certain point, then the cam-roller will be brought by the bell-crank out of the position for meeting the cam, which, consequently, will cause the inlet valve to remain closed during one or more strokes of the piston.

FIG. 136.

The Modern Four-Cycle Throttling Engine.—Figures 137, 138, and 139 are the Plan View, the Front Elevation and the Longitudinal Section of a common type of four-cycle throttling engine, which has been used with particular success for producer-gas. The views represent an engine originally designed by Mr. Max Munzel, late of the firm G. Luther of Braunschweig, Germany, and they afford good illustration for the study of the details of engines of this type.

Fig. 139 shows the main frame and the cylinder jacket to be cast in one piece, and the cylinder bushing to be inserted in the jacket-casting and held in place by a flange at the rear end; thus leaving its front end free to expand or contract without throwing strains on to the main casting. The inlet- and exhaust-valves and the igniter are located in a cylinder-head casting that contains the main part of the combustion-chamber, and which is thoroughly water-cooled all around. The exhaust-valve is readily removable by removing the inlet valve-bonnet, and it is seated on a removable steel bushing.

The cam-shaft is driven, at one-half the speed of the main shaft, by means of a pair of spiral gears located in an oil-tight gear-casing, as shown in Figs. 137 and 138, and the governor is also driven by similar means, similarly located. The inlet
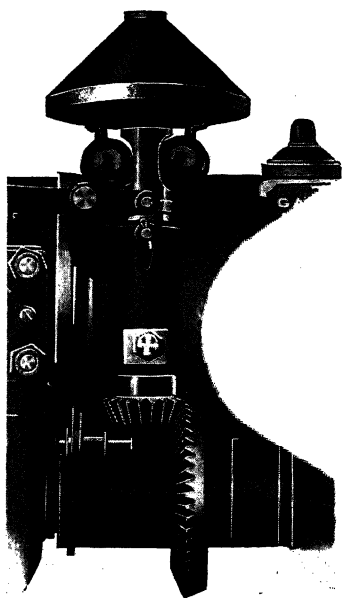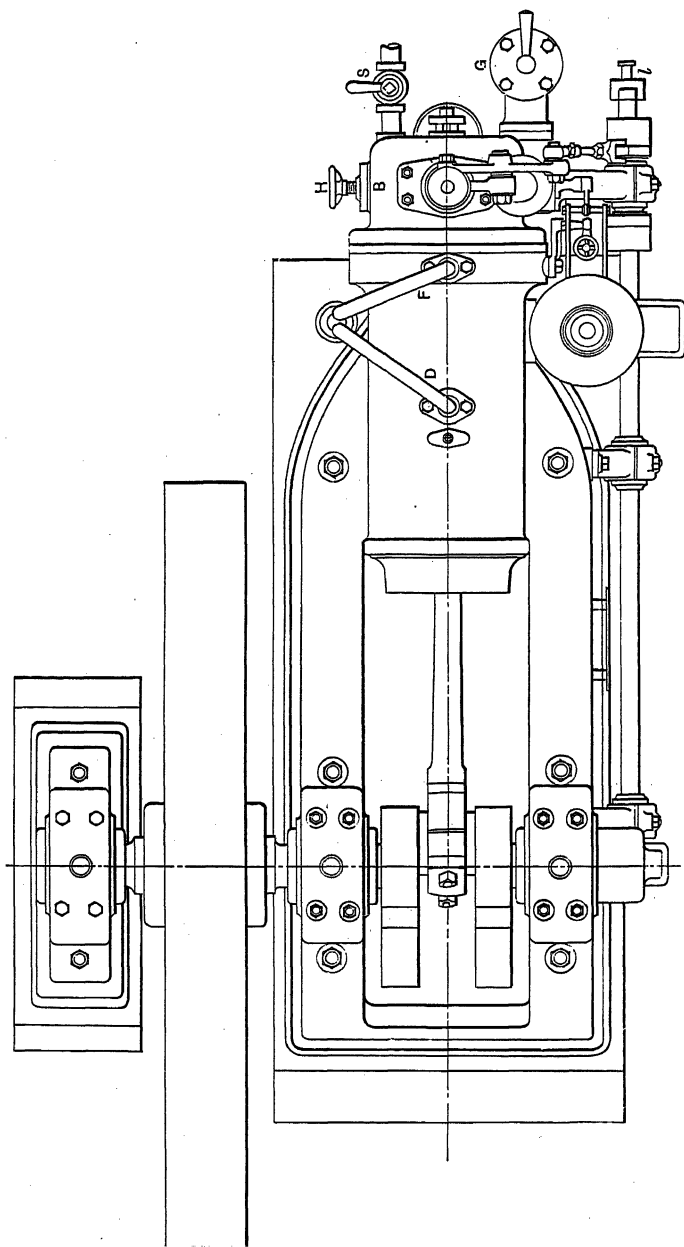
FIG. 137.—Four-Cycle Single-Acting Gas-Engine. Plan View.
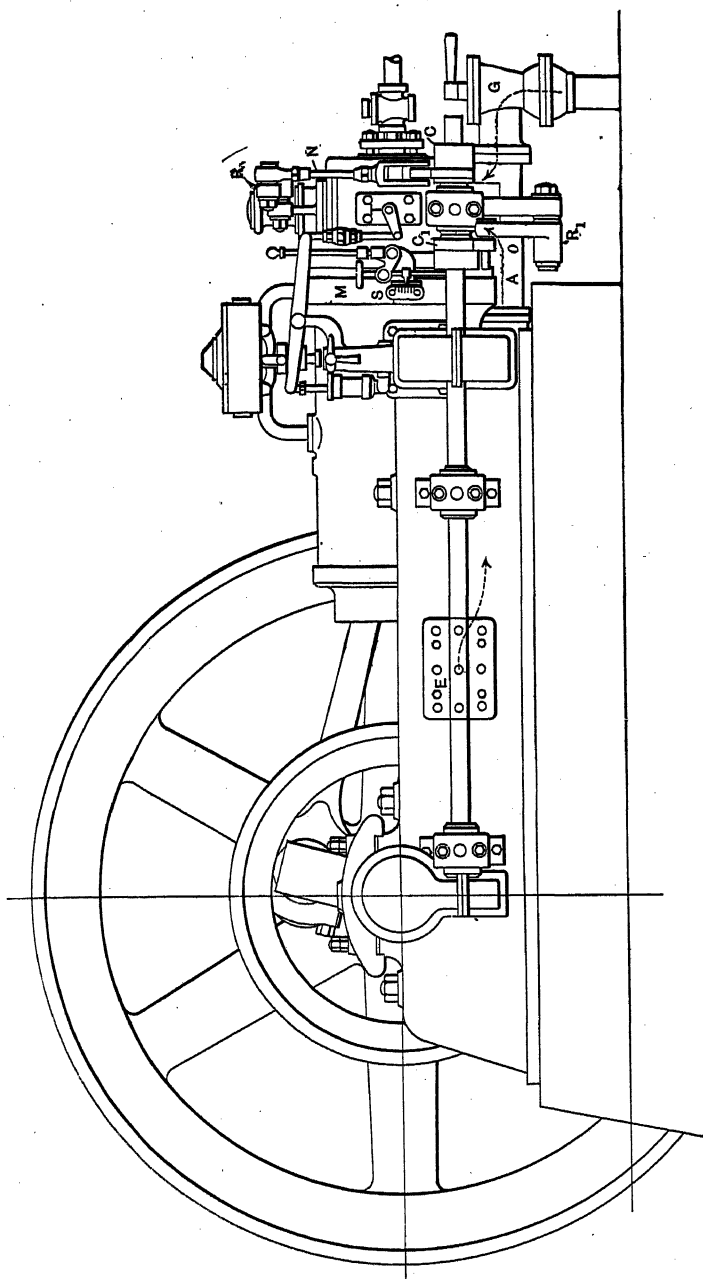
FIG. 138.—Four-Cycle Single-Acting Gas-Engine. Front Elevation.

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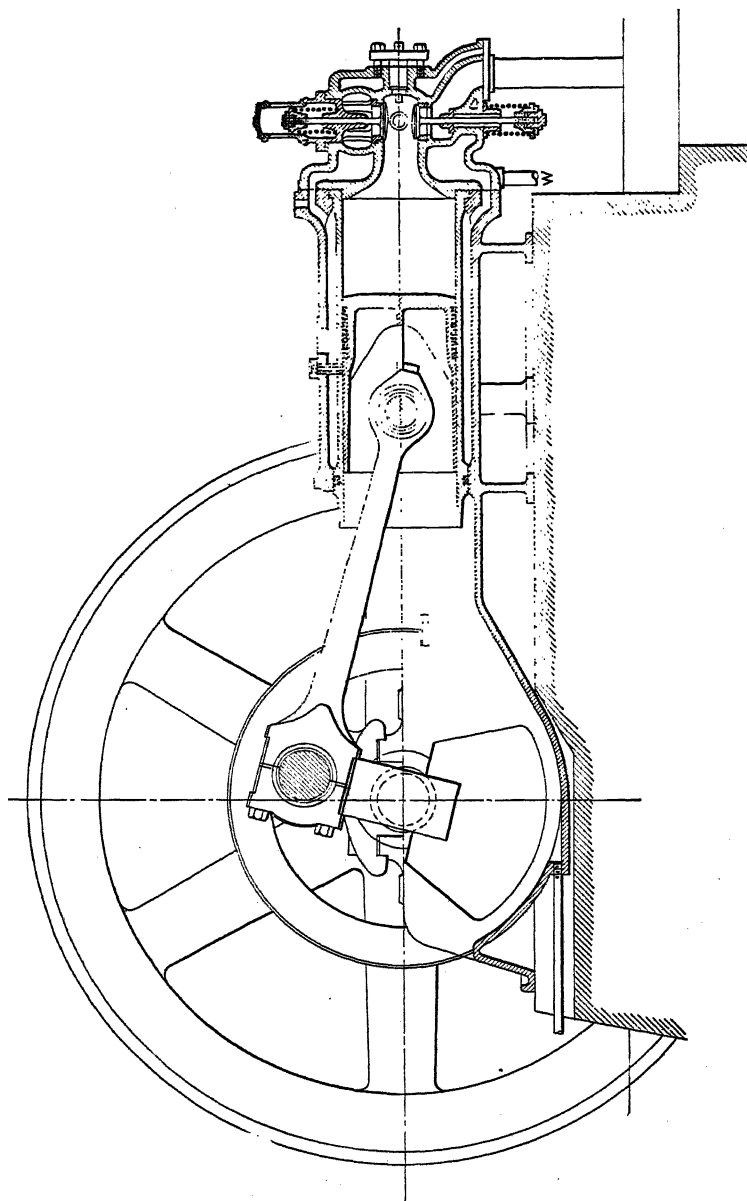

valve is actuated by means of the cam *C*, valve-rocker connection *N*, and valve-rocker *R*. To relieve the compression, at the time of starting the engine, the exhaust cam-roller *o*, can be slid sideways off the regular exhaust-cam, C_1, and on to a smaller starting-cam located at the side of the cam C_1, which gives less than one-half the regular compression-pressure.

The engine draws its charge of air through the frame, which helps to keep the engine cool. The air enters through some holes back of the plate *E* and is drawn in to the mixing-chamber through the pipe *A*, while the gas is drawn through the gas valve *G;* and the gases are mixed immediately below the throttle valve *T*. The proportioning of the mixture is done by opening or closing a throttle-valve located in the air-pipe *A*, by means of the small hand-wheel *M;* and the position of this throttle is indicated by an index running over a scale *s*. By running this index slightly up or down the scale when the engine is operating, and noting the effect the various positions of the throttle will have on the speed, or on the sound of the engine, the best proportioning can readily be ascertained.

The ignition of the charge is effected in this engine by means of the magneto ignition illustrated and described at page 332; the pick-blade lever *L* attaching to the crank *l*, Fig. 137.

The cooling water is admitted to the jacket at a lower point, at *W*, and it is drained from two points *D* and *F* at the top of the cylinder; each drain having a separate regulating-valve. The object with this arrangement is to allow the regulation of the temperature of the water in the combustion-chamber jacket independently of that in the cylinder-jacket; still having both jackets connected.

The starting of an engine of this type is generally accomplished by means of compressed air, which is supplied through the starting air valve *S*, and admitted to the cylinder through a check-valve located back of the bonnet *B*. The hand-wheel *H* is for the purpose of locking the check-valve to its seat during regular running. The position of this valve, relatively to the main valves of the engine, may be seen to better advantage in Fig. 105, page 307, which represents a cross-section through the combustion-

chamber of an engine practically the same as the one being described, excepting that the governor throttle valve is, in Fig. 105, put as close to the inlet-valve as possible. The advantage of this latter arrangement is that the mixing of the air and the gas is effected closer to the inlet-valve, and, hence, the volume of the mixing-chamber is cut down. This feature may be quite desirable when the fuel is of the kind that is apt to cause back-firing into the mixing-chamber.

Manipulation at the Starting of the Four-Cycle Engine.—To prepare for the starting of the engine the main crank must

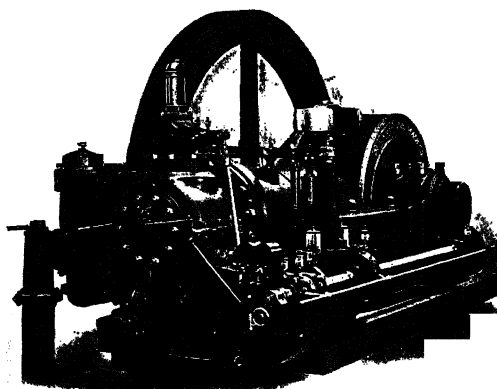



FIG. 140.—Koerting Four-Cycle Engine.

be put in its position for a firing stroke, and a little above the head-end centre; the inlet- and exhaust-valves then being both closed. The hand-wheel *M* is adjusted for a proper mixture, the exhaust cam-roller put on the starting cam, and the hand-wheel *H* turned so as to release the air check-valve. The leads from the magneto are then connected to the spark-plug, and the ignition retarded so as to fire late.

To start the engine the gas is turned on and the air-starting valve *S* is opened to admit air-pressure to the cylinder, and kept open until the crank has started to move, then quickly closed. During the next following forward stroke, the engine will draw its regular charge, which will be fired at the end of the return

compression-stroke. The engine-piston, thus, is called upon to make one exhaust-stroke, one suction-stroke and one compression-stroke due to the impulse from the air-charge. The air-pressure, therefore, is required to be quite high—100 to 150 pounds is variously used, depending on the compression pressure employed.

An alternate method for starting would, of course, be to give the engine several impulses by compressed air, to bring it up to speed before the gas-valve is opened. When the start is made the exhaust cam-roller is put over on the regular cam for normal compression, the ignition is advanced to bring the engine up to normal speed, the lubricators are attended to, and the cooling water for the jackets turned on.

The Koerting Four-Cycle Engine.—An engine of very much the same general construction as the Munzel engine is illustrated in Fig. 140, and the arrangement of its valves is shown in the transverse and longitudinal sections through its combustion-chamber and valve-casings, Figs. 141 and 142.

When engines of this type are connected together to one shaft as twin engines, with a common fuel supply, or when two engines draw from a common gas-main, it will be of advantage to apply in the fuel-supply port of the engine a check-valve that will prevent one engine from drawing the charge back from the other. Such a valve is applied in the Koerting engine and it serves also as a mixing-valve for separating the air and the fuel, until the charge is drawn in by the engine.

The gas and the air arrive at the mixing-valve through the pipes marked, respectively, *G* and *A*, in Fig. 141. When the valve, which covers both the gas and the air ports, is raised, due to the suction of the engine, the gases will flow as indicated by the arrows, and mix in the chamber immediately above the valve. After the suction-stroke is completed, the valve will seat itself and prevent any back flow of the fuel-mixture. In seating, the valve is cushioned by the dashpot formed in the bonnet of the mixing-valve chamber. This dashpot becomes accessible for adjustment by removing the protecting hood placed over it. The governor throttling-valve is, as will be seen, located between the mixing-valve and the main inlet valve.

In engines of this type, the precaution would be justified to apply a relief valve to the mixing-chamber, to insure against breakage liable to occur, due to back-firing into the mixture enclosed between the mixing-valve and the main inlet valve; and it is always a mistake to assume that the channels outside of the

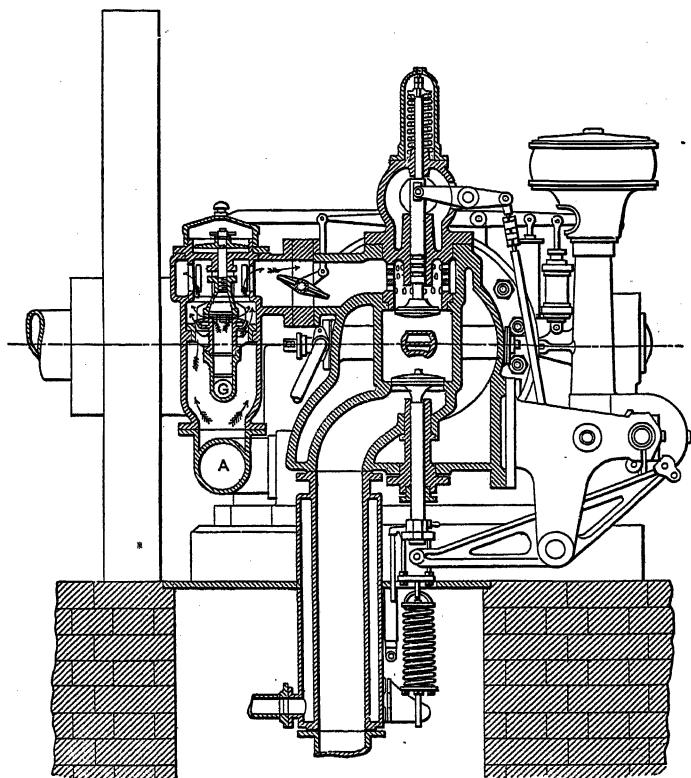



FIG. 141.—Koerting Valve Gear. Transverse Section through Combustion-Chamber.

main valve are not to be counted on to withstand any essential pressure.

Another feature of the Koerting four-cycle engine deserves description. Between the inlet- and the exhaust-valve there projects a box-shaped casting, as seen in Fig. 142, from the cylinderhead to the full depth of the combustion-chamber. This

box is connected to the cooling-water system and serves to prevent the heating of the incoming gases by the exhaust-valve. It has been found that when the inlet- and exhaust-valves are placed close to each other, and so that the charge in passing over the exhaust-valve becomes effectively heated by it, pre-ignitions of fuels liable to such are apt to occur. This is effectively prevented by

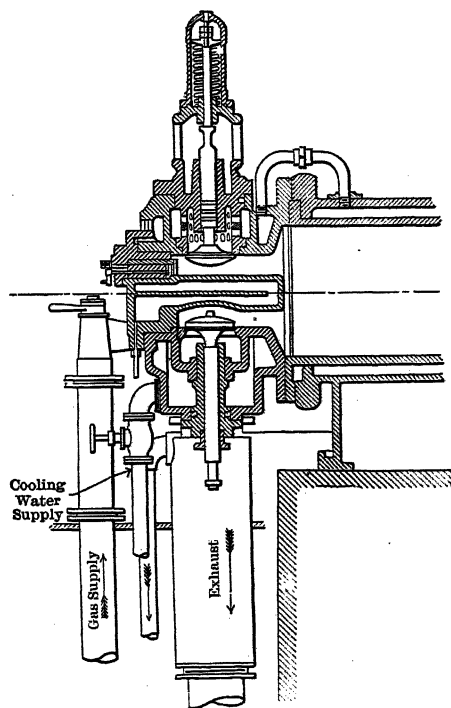



Fig. 142.—Koerting Valve Gear. Longitudinal Section through Cylinder.

the cooler-box applied in this engine. As, further, on account of it the charge will fill the cylinder at a greater density, it serves also to increase, to some extent, the capacity of the engine.

The general arrangement of the valve-gear of the Koerting four-cycle engine is clearly seen by reference to the illustration, Fig. 140, and it is, in its main features, substantially the same as that of the Munzel engine, previously described.

The Olds Engine.—The Olds Gas Power Co. of Lansing, Mich., build a double-throw-crank engine of the same general type as the ones just described; and, as far as its main parts are concerned, also of very much the same construction. Its valve mechanism is, however, materially different from those of the Munzel or Koerting engines, and will warrant a special study.

The valve gear is shown, fully, in detail in the illustrations, Fig. 143, which is a section through the combustion-chamber and valve-casings, and Fig. 144, which is a side elevation of the head end of the cylinder, looking from the valve-gear side. Referring to Fig. 143, it will be seen that the motion for the admission valve is derived from the cam-roller *R*, through the valve-lever connection *C* and valve-lever *L*. As in the Deutz valve gear described previously, this lever is not hinged to a fixed fulcrum, but the fulcrum, against which the lever is forced in opening the valve, consists of a movable block *B*, the position of which is controlled by the governor. The governor-lever *G*, it will be seen, controls the motion of the arm *A* to which is linked the block *B*. Hence, at a slow speed of the engine the fulcrum-block will be moved outward, away from the valve-end of the lever, causing a high lift to be given to the admission valve, and at a high speed the block will be moved inward, causing the lift of the valve to be reduced.

The admission valve consists of the main inlet valve and of the gas valve. The latter being a double-ported piston-valve sleeve, which is secured concentrically on the inlet valve-stem. The inlet valve and the gas valve will thus move together as one, but the gas valve, having a certain amount of lap, will not open the gas port before the main valve has been open some little time for the admission of air, only, from the air-passage, and it will close the gas port before the main valve closes against the admittance of air. The object with this arrangement is to scavenge the valve-casing from any explosive mixture, which precaution makes the engine safer against back-firing into the valve-chamber. Furthermore, the regulation becomes somewhat on the order of a constant-quantity regulation, which appears desirable for certain fuels.

It will readily be seen if we assume that the admission valve, at a very light load, is lifted hardly enough to uncover the lap of the gas valve that, then, air only would be admitted during the suction stroke. By opening the valves a little further principally air, and only a little gas, will be admitted; and so forth. And at the full lift of the valves a normal charge will result. Showing that, for decreasing loads and decreasing valve-lifts, the charge will grow more diluted. Of course, the charge will also become of less density.

The gas stop-valve *V*, it will be seen, consists of a cylindrical sleeve having a port cut through it at one side, by which, by revolving it, the gas may be throttled more or less, or cut off entirely. A more permanent proportioning of the air in the mixture may be made by a butterfly valve which is applied in the air passage.

The valve gear illustrated, which is for a cylinder somewhat more than 21 inches in diameter, is equipped with a water-cooled exhaust valve; the cooling water being supplied at *a*, and drained through the small central tube, inside the valve-stem, which terminates at *b*.

The illustrations, Fig. 143 and Fig. 144, show the engine to be fitted with an automatic air-starting device. This feature is shown at *D*, Fig. 144, and in detail in Fig. 145.

Instead of operating the air-starting valve by hand, as described at page 363, the valve can, of course, be automatically opened and closed. This is done in the present engine by means of the small bell-crank and cam-roller shown at *E*, Fig. 143; the cam-roller being actuated by a special starting cam *e* on the end of the cam-shaft. After the engine has been put in a position for starting, which is with the crank a few degrees above the head-end centre and with the inlet- and exhaust-valves closed, the small cam *S* on the air-starting handle, Fig. 145, is forced against the valve-stem, which has for effect to open the valve. At this time the cam *e* is underneath the cam-roller *r*, but at the proper time for the closing of the air valve the roller will ride off the cam and allow the valve to close. The air valve may thus be worked continuously by the air-starting cam *e*, for several turns of the

FIG. 143.—Olds Valve Gear. Section through Combustion-Chamber.

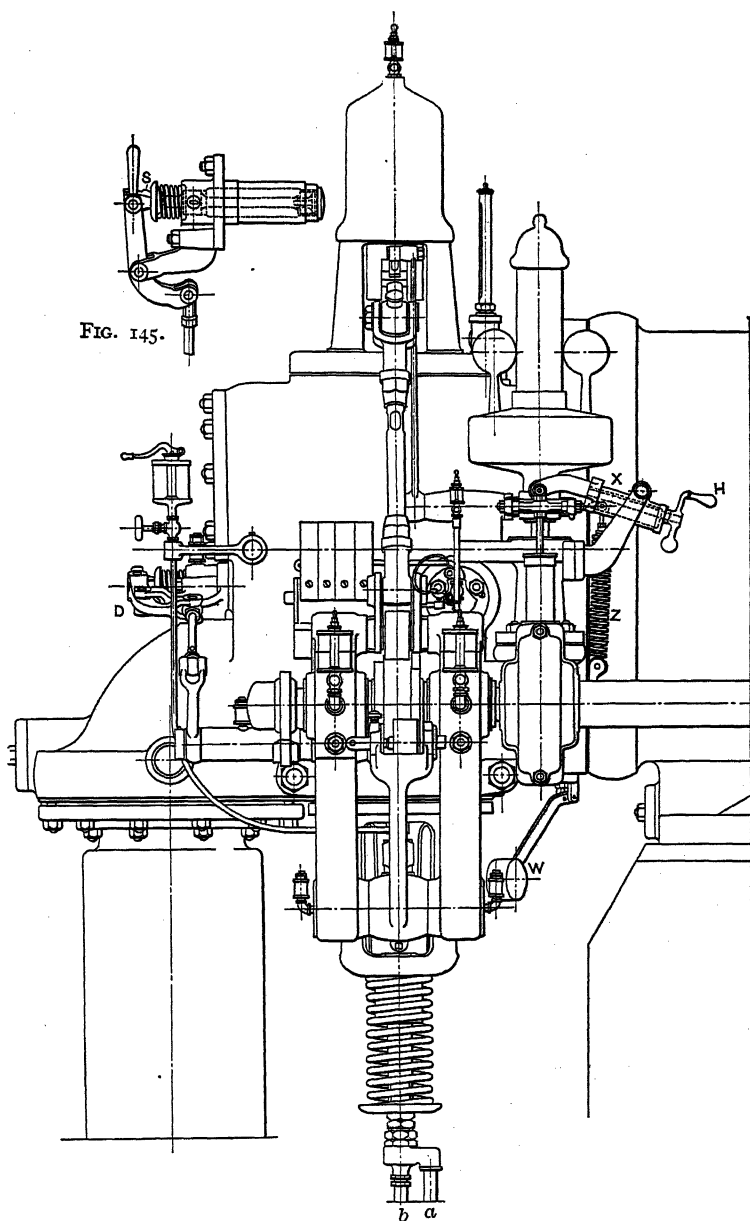



FIG. 145.

FIG. 144.—Olds Valve Gear. Front Elevation.

engine, until the cam *S* on the starting handle is withdrawn from contact with the end of the valve-stem, by the shifting of the handle to a position in a right angle to that it occupies in the figure. With the cam *S* withdrawn, the starting-valve will be closed and the roller *r* hang fire of the starting cam.

The combustion-chamber is, at the bottom, provided with a small drain and relief valve, which is counterweighted by the weight *W*.

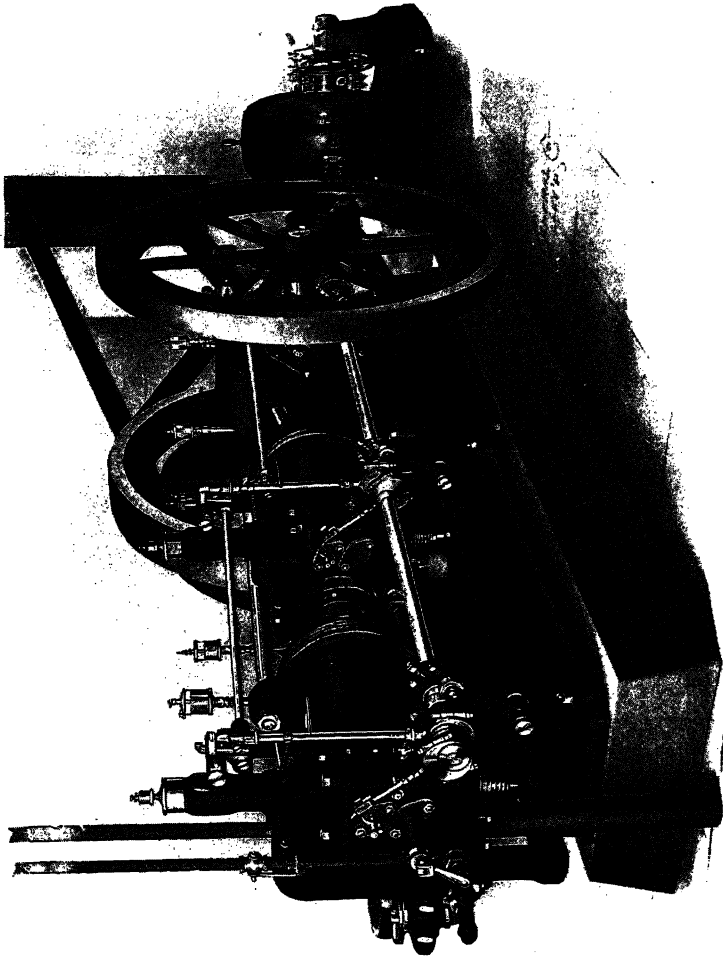
By means of the handle *H*, the speed of the engine may be varied within some few turns. The leverage by which the spring *z* acts on the governor-sleeve can, namely, be changed, within the limits of the slide *X*.

Multiple-Cylinder Engines.—Open-end cylinders below 21 inches are not generally equipped with water-cooled pistons, as the atmosphere, due to the motion of the piston, will have adequate access for cooling it. Neither is, as a rule, the exhaust-valve of cylinders below that size water-cooled. An engine of a single 21-inch cylinder, running at normal speed, will on producer-gas develop, on an average, 125 B.H.P.; or if constructed as a two-cylinder engine twice this power will be obtained.

For installations below 250 to 300 B.H.P., therefore, the single-acting four-cycle engine becomes, on account of its simplicity, a very serviceable type. Two-cylinder engines are by some builders arranged as twin engines; others prefer the tandem arrangement, but as far as the regulation or the required weight of wheel is concerned one type is as favorable as the other. The space available for an installation may, however, in a great measure become a determining factor with regard to the type most suitable in any specific case.

As to twin engines; any of the four-cycle engine types described in the preceding may, when occasion requires, be arranged as such, simply by coupling together two engines on one shaft having both cranks in line.

The Jacobson Tandem Engine.—Fig. 146 illustrates a type of tandem engine built by the Jacobson Machine Co., of Warren, Pa. The view, which is looking from the valve-gear side of the engine, shows the valve-gear lay-shaft to be driven by a pair of spiral

gears from the main engine-shaft, and its speed is, of course, one-half the speed of the latter. The exhaust-valves are, as the figure plainly shows, operated by cams, while the inlet-valves are actuated by means of eccentrics.

The valve-gear is shown in detail in the sectional view, Fig. 147, and it will be observed that the regulation is effected by means of a release gear by which the inlet-valve is disengaged and closed at varying points of the stroke, to suit the load. To

some extent the cylinders are scavenged with pure air in order to prevent, as much as possible, any back-firing due to slow-burning mixtures.

The operation of the gear is as follows:

The inlet valve *I* is opened by means of the rocker *R* when the pick-blade *P* on the end of the eccentric rod *E*, lifts the end of the rocker-arm, and the valve becomes opened more and more, until the roller *D* rides up on the trip-block *T*, the position of which is controlled by the governor. It is evident that the further the trip-block is moved to the right the earlier in the stroke the valve will trip.

As soon as the main inlet valve opens, air will be admitted to the cylinder from the air port, for scavenging purposes. The gas valve *G* has a sliding fit on the main valve-stem and is held closed by the spring *Z*, until the collar *C*, which is solid on the valve-spindle, moving down, forces the valve to open against the spring-pressure back of it. The position in which the valves are drawn is such, it will be observed, that the main valve is opened to some extent for the admission of air, and the collar *C* is just touching the hub of the gas valve to effect its opening, in unison with the main valve, as the latter is further opened. *S* is the main valve-spring, the function of which is to close the main inlet valve, and to hold it closed against the partial vacuum that will be created in the cylinder at light loads.

For the cushioning of the valves, in seating, a cushioning-pot and piston are provided at *O*. Means are also provided in the gas and air ports, whereby the proportioning of the air and gas may be regulated to suit any kind of fuel that may be used.

The Premier Engine.—A well-known English tandem engine is, in sizes above 150 horse-power, built by the Premier Gas Engine Co., at Sandiacre, Nottingham, Eng. The arrangement of its cylinders and general construction is the same as that of the engine described above, excepting that the Premier engine is positively scavenged. The main crosshead, working in a bored guide-cylinder, serves the double purpose of guiding the end of the piston-rod and, as a piston, for compressing the scavenging air.

A test of an engine of this make running on Mond gas is reported by Mr. Humphrey in the Proceedings of the Institution of Mechanical Engineers, vol. 1901, page 79, and the principal figures of this test may be found in Table XXXI. The main sizes of the engine are: the working cylinders 28⅛ inch diameter

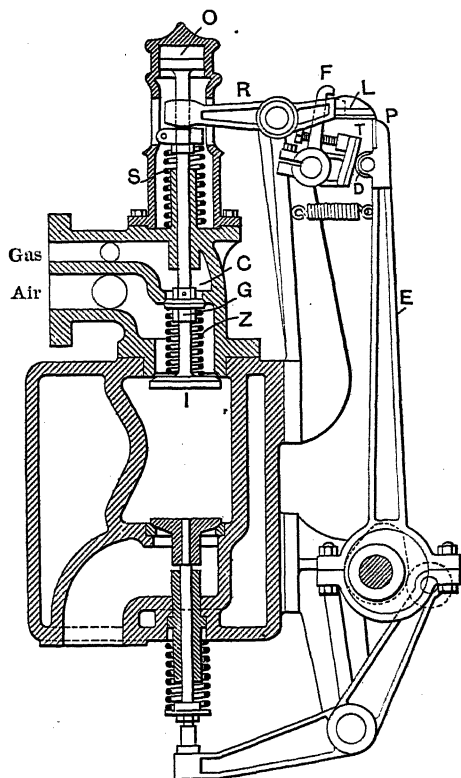
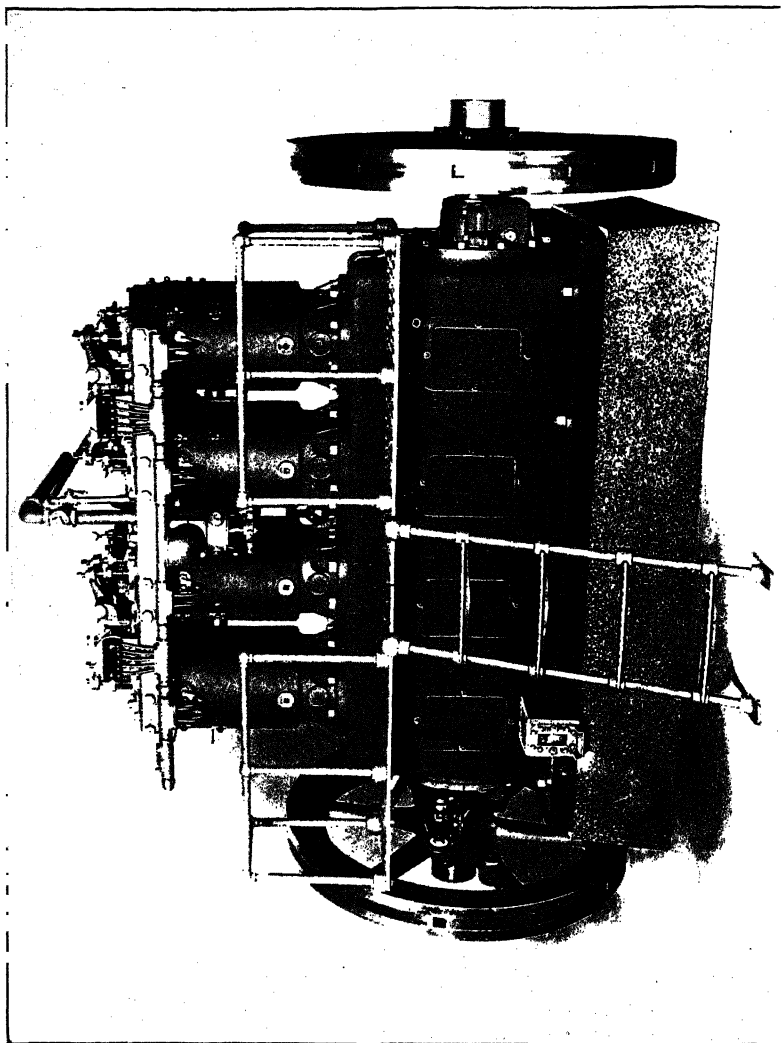



FIG. 147.—Jacobson Valve Gear.

x 30 inch stroke, the pump cylinder 43½ inch diameter x 30 inch stroke. The pistons are water-cooled, and to this fact, as well as to the positive scavenging of the cylinders, may be ascribed the very high mean effective pressure obtained. The test referred to reports an average mean effective pressure of 107 pounds. Figured from the power developed and the displacement volume

FIG Four-C Engin

of the pistons, the average mean effective pressure will be 81 pounds; to which, however, there should be added the pressure corresponding to the negative work of the pump piston to get the actual effective pressure in the working cylinders.

An average mean effective pressure as high as 81 pounds is not generally obtained with 144 B.T.U. gas in the non-scavenging engine, excepting for short periods of particularly suitable and steady load.

Vertical Multiple-Cylinder Engines.—In Figs. 148 and 148*a* is illustrated a type of multiple-cylinder gas-engine which is extensively installed, in units from 30 to 400 horse-power, for lighting as well as for motive-power purposes. This engine-type is built of two, three, or four cylinders, hence, with only a few cylinder sizes, a great variance of power can be obtained; and, it being of a very compact arrangement, it is particularly suitable when the space for an installation is limited.

From the illustration, Fig. 148, which is a reproduction of an engine built by the Bruce-Macbeth Engine Co., of Cleveland, O., in units of 100 horse-power up, a fair idea may be gained of the general construction of an engine of this type. The gas-supply elbow, the governor throttling-valve, and the mixture-supply chamber communicating to the various cylinders are plainly seen in front of the cylinders, whereas, correspondingly to the supply chamber, in front, there is, back of the cylinders, an exhaust manifold, which leads directly from the exhaust-valves, through a muffler, to the atmosphere. The inlet- and exhaust-valves are actuated by means of four cam-rockers, and cams on the cam-shafts located between each pair of cylinders. The transmission for motion from the main engine-shaft to the cam-shafts is plainly seen in Fig. 148*a*, which is a section through one cylinder and the crank-casing. The same figure shows also the governor, which controls the throttling valve by means of a long lever fulcrumed between the two middle cylinders, as well as the exhaust manifold referred to above.

At the left side of the crank-chamber, Fig. 148, there is shown a small generator driven by means of a belt from the fly-wheel hub, and it supplies the current for the high-tension spark-plugs

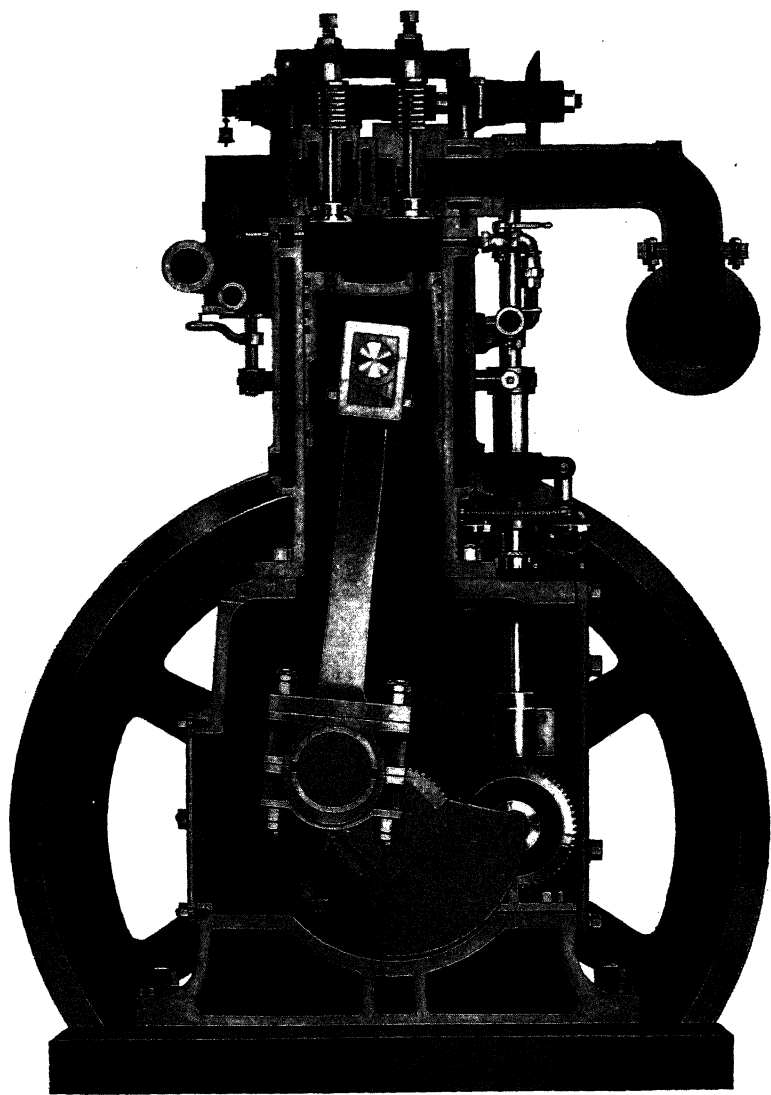



FIG. 148*a*.—Bruce-Macbeth Four-Cylinder Engine. Section.

fitted, in duplicate, to each cylinder. One set of spark-plugs may be furnished with current from another source than that of the generator shown, so as to make the proper ignition doubly assured.

One feature of the four-cylinder four-cycle engine, which recommends itself very much for large engines, is that its starting becomes very convenient. One air starting-valve is furnished for each combustion-chamber, the operation of which is effected by cams on the main cam-shaft, and so timed that the air ad-

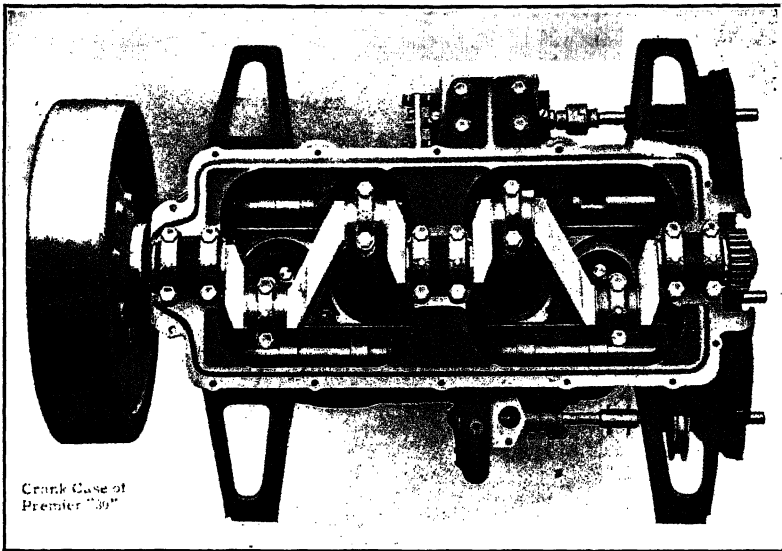



FIG. 149*a*.

mission corresponds with periods when both main valves are closed. When compressed air is turned on to the starting-valves, they will remain closed, excepting when the cam-throws are in such position as to engage with the valve-stems and force the valves open, each one in turn. The engine is, thus, conveniently started, from any position, simply by turning on the compressed air.

The Automobile Engine.—A modern four-cylinder type of automobile motor of high power is illustrated in Figs. 149*a* and 149*b*. The design is of the Premier Motor M'f'g Co. of Indianapolis,

Ind., and it represents the latest improvements in the American and European automobile motor construction.

Fig. 149*a* is an inverted plan view of the engine, with the lower part of the casing removed, so as to show plainly the arrangement of the crank-journals and the location of the cam-shafts which actuate the admission and discharge valves. The cam-shafts

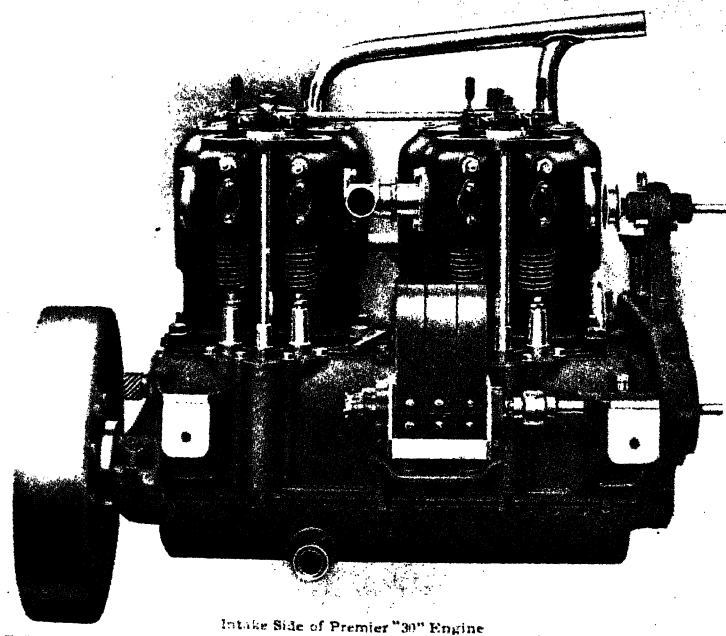



FIG. 149*b*.

are driven, it will be seen, from the end of the crank-shaft by means of spur-gears, which are arranged in a general gear-box at the front end of the engine. In the same gear-box are gears provided also for driving the circulating pump and the ignition-magneto. The igniter-spindles are operated from the admission cam-shaft by means of two pairs of spiral gears, while the lubricator is driven by means of spiral gears from the discharge cam-shaft.

Fig. 149*b* is a view looking from the admission side of the engine. It shows the arrangement of the four cylinders with their admission valve-casings, and the igniter-spindles coming up from the crank-case; each spindle operating the igniters for two cylinders. The magneto, of the Bosch type, is secured to the crank-case, in front of the cylinders, and its armature-spindle is driven, through a universal coupling, by means of a spur-gear in the general gear-box.

Figs. 150*a* and 150*b*, respectively a combined half longitudinal section and half front elevation, and a cross-section through one cylinder and crank-casing, show the detailed construction of the engines.

The cylinders, of a bore of 4½ inches, are cast in pairs, with the inlet and exhaust valve-chambers arranged on opposite sides of them. They are water-cooled as far down as the working space, and both valve-chambers are water-cooled, completely, all around the valve-seats. The cylinders and pistons are, of course, ground true, and the end of the piston is given a very slight taper, to allow for the expansion of the outside material when heated. Four eccentric piston spring-rings are used. They are ground true, and each pair fitted, freely, in each of the two grooves turned and ground near the end of the piston. Several oil grooves are, properly, cut in the lower part of the piston to retain and distribute the lubricant.

The crank-case, of close-grained cast iron, is made as light as possible, and the crank-case cover, to save weight, is made of sheet steel, principally. Aluminum cases have been used to some advantage, as far as the reducing of weight is concerned, but this metal does hardly seem to possess the necessary strength and stiffness to give the same rigidity to the framework of the engine as cast iron, and so far as durability of machined surfaces and threads are concerned it is considerably the inferior.

The crank-shaft is made of suitable drop-forged steel, and it is journaled in Parsons white-brass bearing shells, which metal, although it is of considerable stiffness, possesses the necessary softness for making a suitable journal.

The connecting-rods are made as light as possible by having

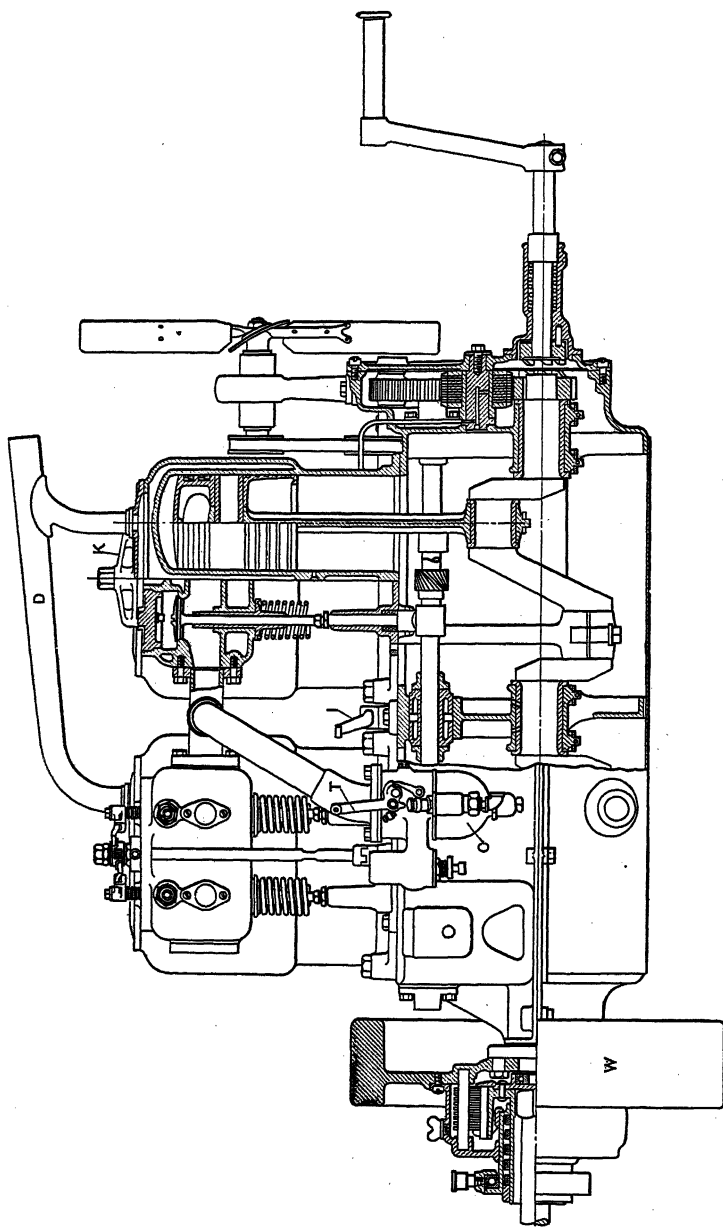



FIG. 150*a*.—Automobile Engine. Elevation and Longitudinal Section.

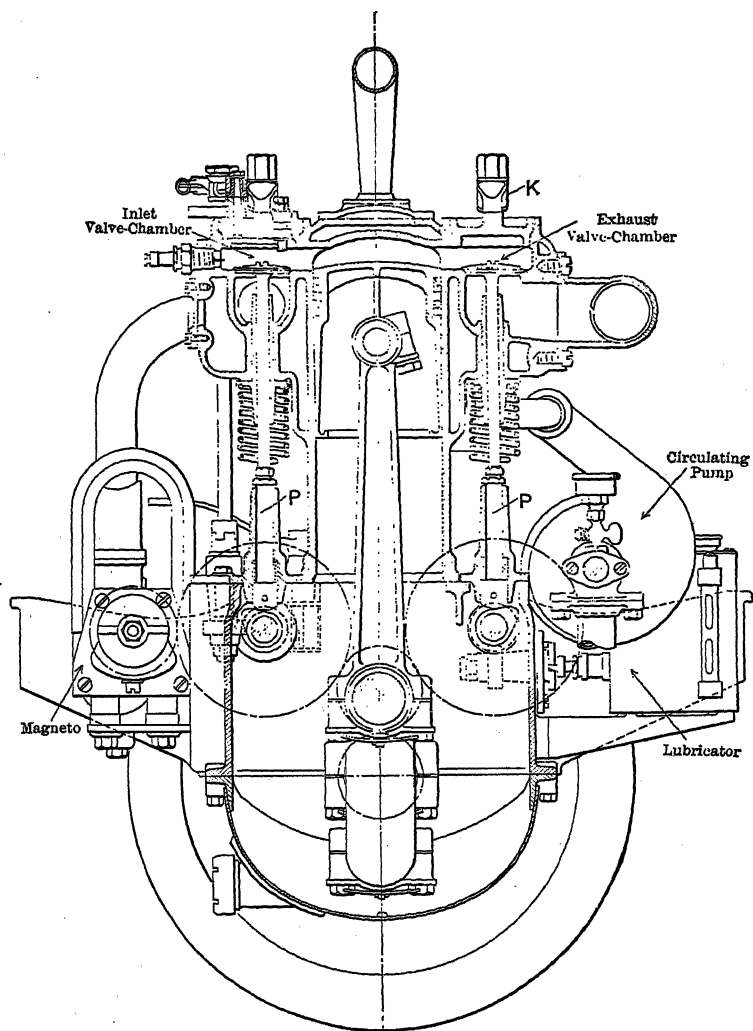



FIG. 150*b*.—Automobile Engine. Cross-Section.

the body of the rod formed in a channel-section; the metal, thus, well distributed for obtaining the greatest stiffness of the rod.

The valves are of nickel steel, and guided by means of their stems in long bushings so that they must always seat true. To effect the lifting of the valves, the push-rods, *P*, simply butt against the ends of the valve-stems, and the timing of the lift may, to some extent, be adjusted, by means of a locked adjusting screw in the end of the push-rod. Each pair of covers for the valve-casings are held closed by means of a clamp, *K*, which is tightened down on the covers by a heavy bolt. Thus, by unfastening four bolts the eight valve-covers become free for removal. In the covers of the admission valve-casings, as housings, there are arranged the igniter-rods of the low-tension make-and-break igniters. These covers serve, therefore, also the purpose of ordinary make-and-break spark-plugs, while high-tension spark-plugs are screwed into the side of the admission valve-casings. The two systems of ignition are applied to the engine, so that in case of failure of one the other can be resorted to.

The carbureter, shown at *C*, Fig. 150*a*, is equipped with a throttle valve, by means of which, in combination with the timing of the spark, the speed and power of the motor is controlled. The throttle-lever, *T*, and the igniter-lever, *I*, are controlled by means of throttle- and spark-levers in the steering wheel. The latter are adjusted over a stationary quadrant in the wheel.

The cooling system employed in connection with these engines is in principle the same as that described at page 328. The hot water discharge pipe from the top of the cylinders to the radiator is shown at *D*.

A fan for impelling an air-current through the cells of the radiator, which is placed directly in front of the engine, is shown in Fig. 150*a*, and it is driven by means of a belt from the circulating-pump shaft.

A forced-feed lubricator is located on the exhaust side of the motor, in close proximity to the exhaust-pipe, which thus keeps the lubricant, at all times, at an even and suitable temperature for efficient lubrication. The lubricator is, as has been explained, driven by means of a pair of spiral gears from the discharge cam-

shaft. The oil is fed to the cylinder at a point between the upper and lower set of spring-rings when the piston is at its lower centre.

The make-and-break igniter mechanism, shown for one pair of cylinders in Fig. 149*c*, consists of two small cams secured to the end of the igniter spindle; actuating, each, one of the two igniter cam-levers. The cam-levers are not secured directly to the movable electrodes, but the latter, which are held closed independently by springs, are actuated through forked arms. The cam-levers have ample clearance between the lugs of the

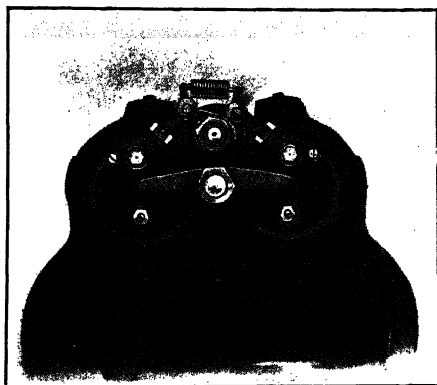



FIG. 149*c*.

forks to allow the igniter to close positively and to be free of any vibrations that may be set up in the levers.

A Weston type multiple-disc clutch, *W*, is employed for clutching in or releasing the motor from the transmission shaft. This clutch consists of a housing attached to the fly-wheel of the motor and of a hub keyed to the transmission shaft. Between these two there are a series of disks, every second of which is fastened to the housing and every second to the hub. By means of a sliding collar, the disks can be squeezed together, tightly, which, due to the friction between the various disks, will cause the transmission shaft and the motor-shaft to revolve together. By varying the pressure with which the disks are pressed together the latter will slip relatively to each other, more or less, thus

varying the speed of the transmission shaft. The clutching in of the motor, by means of this clutch, will always be gradual, and, thus, any sudden shocks in the shafts and gearing are avoided.

The engine is secured solidly to the frame of the vehicle by means of brackets seen, in Figs. 149*a* and *b*, to extend from the crank-case, and in front of it is installed the radiator, which forms the front end of the bonnet enclosing the motor; as seen in Fig. 151. The latter figure shows also the various controlling levers, and those for shifting the clutch-collar and transmission-gears. The figure illustrates, besides, in a striking manner, the relatively small space occupied by a 45-horse-power motor, together with its auxiliaries, as installed in a touring car.

Kerosene and Oil Engines.—The Hornsby-Akroyd Oil Engine— In the Hornsby-Akroyd oil engine, shown in a longitudinal section in Fig. 152, the air only is admitted to the cylinder during the suction stroke, and the fuel-oil is by means of the oil-pump, introduced into the vaporization- and combustion-chamber, *V*, at the time the engine piston commences to draw its air. In this chamber, which is, as seen, separated from the cylinder by a contracted passage, *P*, the fuel is vaporized and mixed with the neutrals remaining from the preceding stroke. When the air-charge, which in the meantime is being compressed in the cylinder, finally, toward the end of the compression stroke, is forced into the vaporization-chamber and mixed with the fuel, the charge will ignite due to the heat of the vaporization-chamber. The expansion stroke and the discharge of the burned gases follow during the next forward- and return-strokes. The engine operates, thus, according to the Otto four-stroke cycle, and with self-ignition.

The air-inlet and exhaust valve-chambers are located side by side back of the cylinder, as viewed in Fig. 152, and stand in communication with the cylinder through the port *A*. The valves are operated by means of the valve-rockers, rollers and cams seen below and in the front of the cylinder, in Fig. 153.

From the latter illustration a fair idea of the general construction of the engine and of many of its details can be obtained.

The oil-pump is operated from the air-inlet valve-lever; the

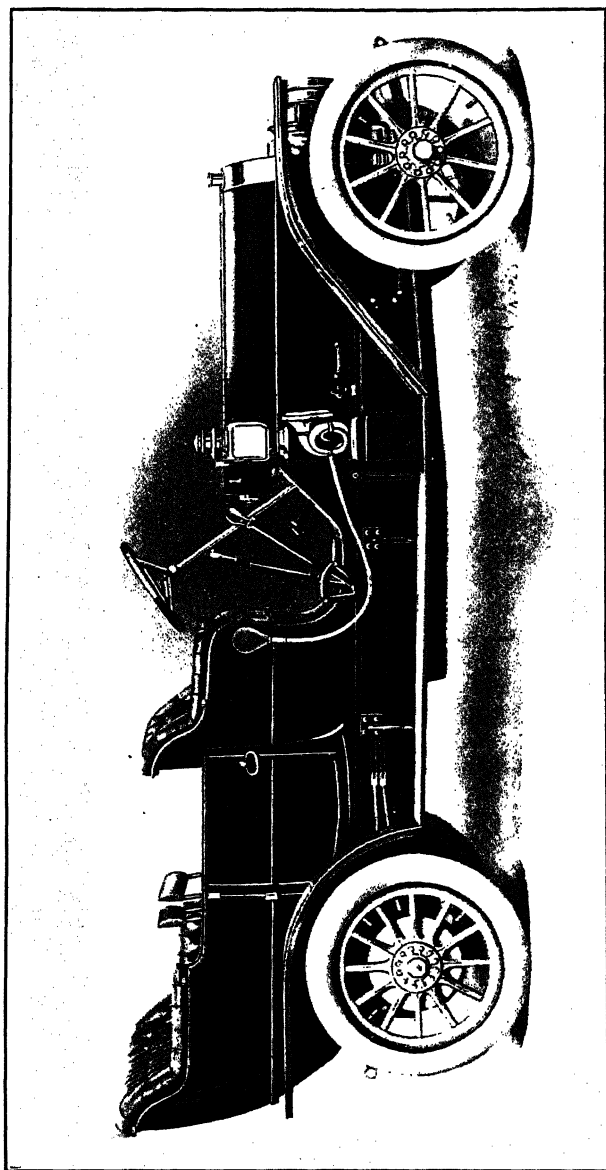



Fig. 151.

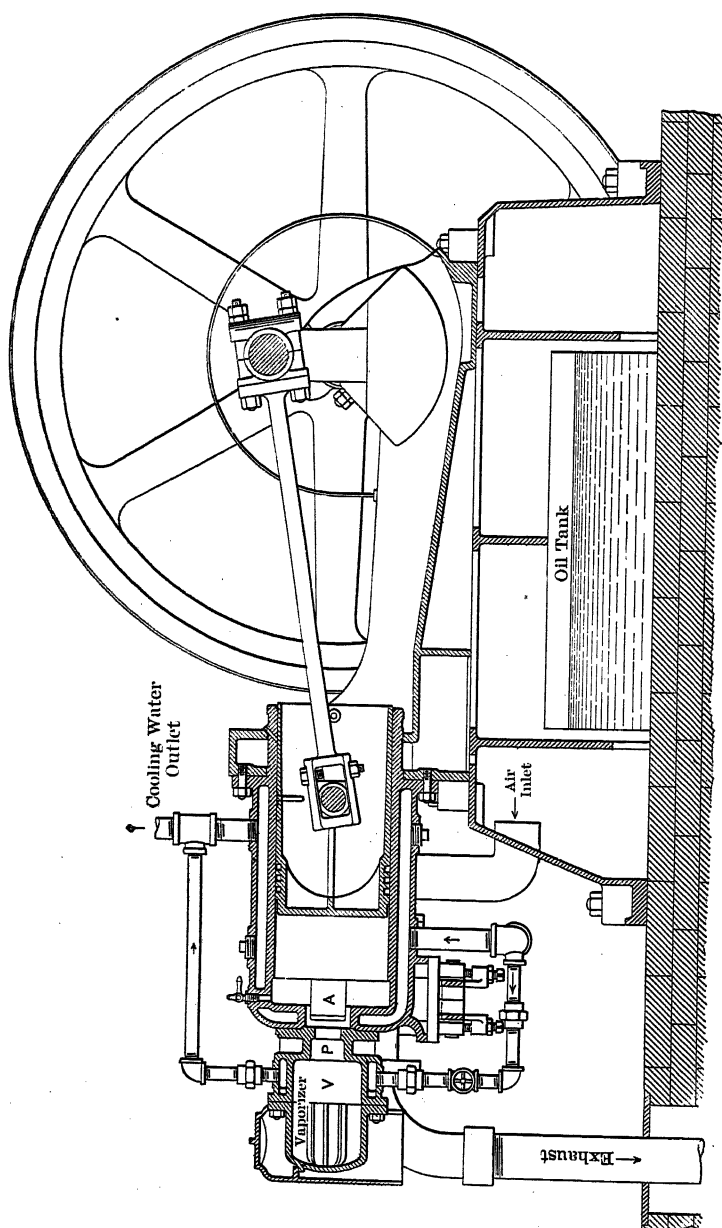



Fig. 152.—Section through Hornsby-Akroyd Oil Engine.

fuel, thus, being forced in to the vaporizer at the time the air valve is being opened for the admission of air to the cylinder, and the amount of oil introduced is controlled by the governor.

Before the engine can be started, the vaporization-chamber must be heated to the required temperature to cause the ignition of the first charge. For the purpose, there is used a torch which is applied underneath the vaporizer and inside the hood which forms the protection for the latter.

The Diesel Oil Engine.—The Diesel engine operates, as has

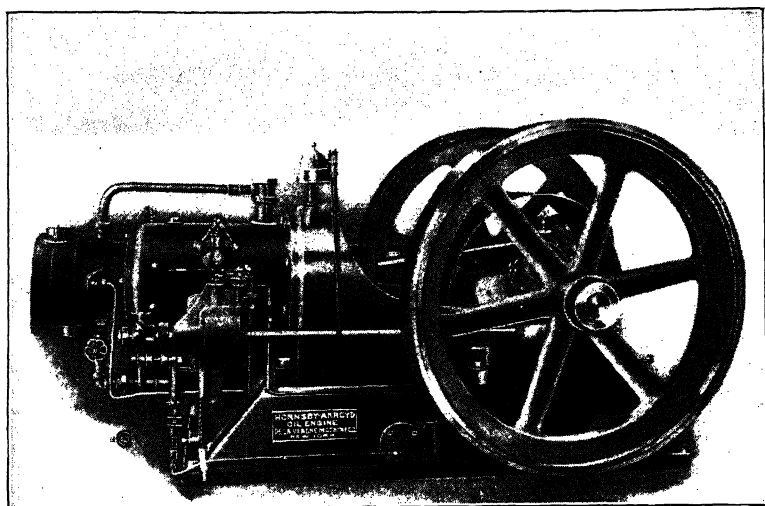



FIG. 153.—Hornsby-Akroyd Oil Engine.

been explained in the preceding, page 24, according to the Brayton or Diesel cycle. It is built commonly in units of one, two-, three-, or four-cylinder machines and of powers of from 25 to 400 brake horse-power.

The engine operates on any petroleum fuel, crude or refined, whose gravity is not less than 19° Baumé. Fuels of 30° Baumé gravity, with a flash-point of 140° to 240° F., are particularly suitable.

Fig. 155 is a cross-section through one cylinder and the crank-case, and it shows plainly the detailed construction of the engine.

Fig. 154.—Installation of Diesel Three-Cylinder Engine.

The air-inlet valve, the exhaust valve, and the fuel valve are as noted in the figure. Each of the above valves is driven by separate cams and cam-levers located inside the crank-casing, and the timing of the valves is according to the diagram Fig. 156.

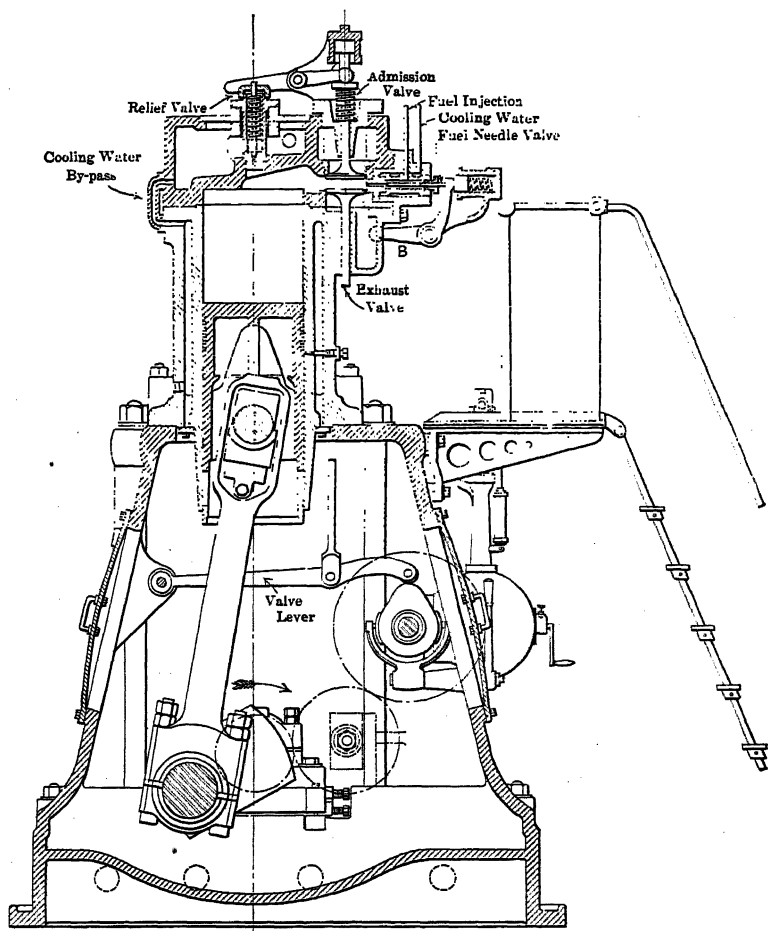



FIG. 155.—Transverse Section through Diesel Oil Engine.

The fuel valve consists of a needle-valve which seats against a steel bushing provided with an admission port of suitable size, through which the liquid fuel-oil is injected by means of an

atomizing jet of air under high pressure. To the fuel-valve casing there is connected, as shown in the figure, the fuel-injection pipe and a water pipe for the cooling of the fuel-valve.

If the operation of the engine be followed through one cycle (consisting of four strokes), it will be as follows: The cycle begins with the suction stroke and the piston on the top centre, and while the piston descends pure air will be admitted to the cylinder, which on the upward stroke will be compressed into the combustion-space to a pressure of approximately 500 pounds. The adiabatic compression of air to 500 pounds will cause its temperature to rise from, say, 60° F. to 930° (computed according to formula 33*a*). Practically, due to absorbed heat, the temperature will become close to 1,100° F., which, of course, is high enough for igniting spontaneously any crude or refined petroleum that may be admitted into the air-volume. When the piston arrives at the top of its compression-stroke, the bell-crank *B* will open the fuel valve, and the fuel is injected, becomes ignited, and burns. The amount of fuel that will be injected is regulated by the governor which is in immediate control of the fuel pump supplying the fuel-charge to the injection pipe, and the supply will, accordingly, be proportionate to the load the engine carries.

After the closing of the fuel value, which occurs when the piston has travelled approximately eight per cent of its downward working stroke, and the fuel is consumed, expansion of the enclosed gases will continue, until, at 90 per cent of the stroke, the exhaust valve opens for release. During the following upward stroke the gases are discharged from the cylinder.

The air for the injection of the fuel is compressed by means of an independent air-compressor to about 1,100 pounds, and it is cooled in a special air tank before it is used.

An installation of a three-cylinder Diesel engine, connected to an electric generator, is represented in Fig. 154. Toward the extreme left of the figure the storage tanks for the compressed injection-air are shown, and the compressor that furnishes the air to the tanks is seen to be belted from the left-hand pulley of the engine.

At full speed of the engine the air charge drawn in to the

cylinder will necessarily be of a pressure somewhat less than the atmospheric, due to resistances in valves and ports; and the charge is compressed to approximately 500 pounds. At a slow speed, in starting the engine, on the other hand, the charge will be of a pressure of practically that of the atmosphere. It has been pointed out that, for high compression, the final pressure of the charge will be considerably higher when the compression begins at the atmosphere than it will when the compression begins only a trifle below the atmosphere, and, in the case of the Diesel engine, the increase in the final pressure at slow speed is so considerable that a relief valve must be employed to release it. This valve, commonly set to lift at a pressure of 800 pounds, is shown in the sectional view of the engine, Fig. 155.

FIG. 156.—Valve-Cam Setting for Diesel Oil Engine.

The Diesel type of engine has, lately, been successfully used for marine purposes. The safety of its fuel and many features about the engine make it seem particularly suitable for such service.

The Double-Acting Four-Cycle Engine.—In this country, with respect to large power-requirements, the double-acting four-cycle engine has lately become the standard engine.

The distribution of the power during the cycle is not in the double-acting single-cylinder engine as favorable as in the tandem or twin single-acting, and it is therefore rarely built as a single unit. Its power-capacity, for the space occupied, is, however, much greater, and on this account the application of the single-acting engine will in the future undoubtedly be limited to power-requirements of approximately 300 to 400 horse-power; and above that power the field for the double-acting tandem, or twin-tandem, will begin.

Fig. 157.—Double-Acting Engine.

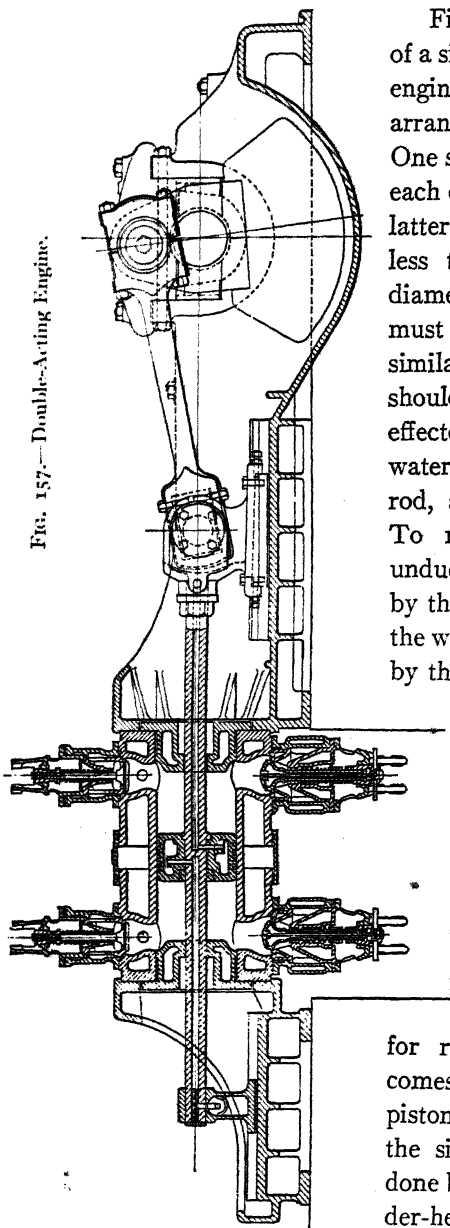



Fig. 157 is a sectional view of a single-cylinder double-acting engine, which shows the general arrangement of its main details. One set of valves is provided for each end of the cylinder, and, the latter not being generally made less than 20 to 22 inches in diameter, the exhaust valves must be water-cooled. For a similar reason the piston also should be cooled, and this is effected by leading the cooling-water through a hollow piston-rod, as explained at page 296. To relieve the cylinder from undue wear liable to be caused by the dead load of the piston, the weight of the latter is carried by the main and the rear cross-heads.

The cylinder-heads are thoroughly cooled and provided with suitable metallic rod-packings, which often are, as in the Schwabe packing, separately cooled.

The piston is secured to the rod solidly, not for removal, and when it becomes necessary to remove the piston from the cylinder it is, in the single-cylinder engine, best done by removing the rear cylinder-head and sliding the piston and rod together out through the

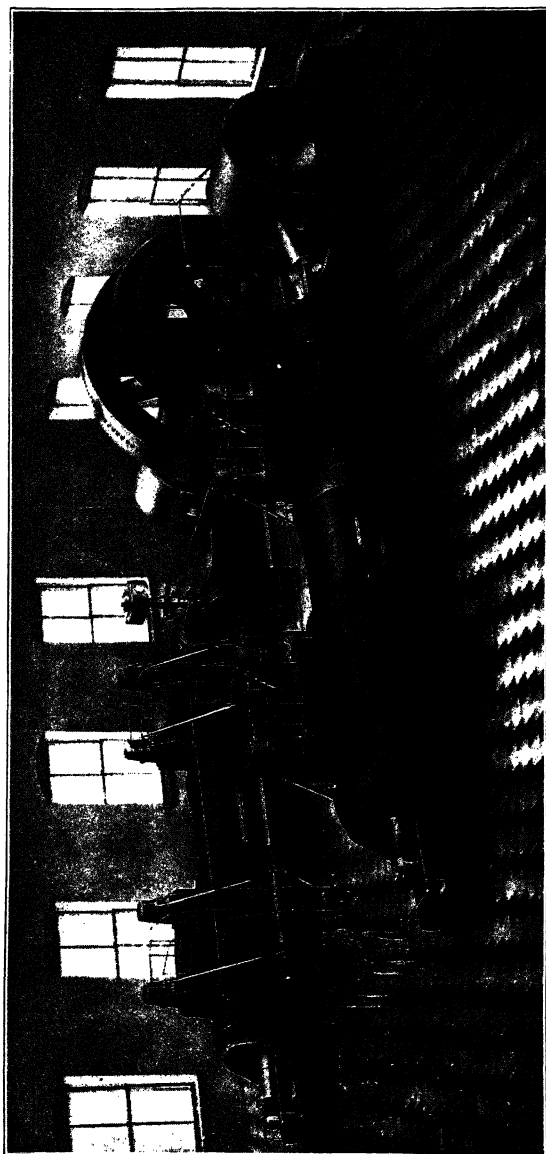



FIG. 158.—Allis-Chalmers Twin Double-Acting Tandem Engine. Direct-Connected to Electric Generator.

rear end of the cylinder. To facilitate this, ample space must be provided at the rear end of the engine. The front piston of a double-acting tandem engine must be removed through the front end of the cylinder, and this is accomplished by removing the main crosshead and front head, and, with the crank and connecting-rod in such a position as not to interfere with the piston-rod, the piston may be slid through to the front.

A double-acting tandem engine, direct-connected to a generator is represented in a plan view and in elevation, in Figs. 159 and 160. It is essentially the same engine as the single-cylinder engine, Fig. 157, with the addition of the rear cylinder and an intermediate crosshead and guide. The piston-rods are coupled together at the intermediate crosshead, and the rods and pistons are removed, one through the front and one through the rear end of the front and rear cylinders.

The cooling-water for the pistons

IG. 159.—I r-C Double-A dem Gas gin Direct-Co Electric C eral Plan Vie

FIG. 160.—Four-Cycle Double-Acting Tandem Gas-Engine. Front Elevation.

is sometimes admitted at the intermediate crosshead, for both pistons; one stream going to the front, to be drained from the main crosshead, while the other goes to the rear, to be drained from the rear crosshead. Sometimes the cooling-water is admitted at the main crosshead, flowing through one after the other of the cylinders and drained from the rear crosshead. The latter arrangement will cause the water going through the rear piston to be somewhat hotter than that going through the front one; and this, it is claimed, will be of disadvantage. The temperature-difference between the gases and the water is, however, so considerable that the matter of some degrees higher or lower temperature of the cooling-water will be of minor importance.

Whenever there are sulphurous vapors present in the gas, the water should be run through the rods hot, so as to prevent any water-vapor from condensing in the rod packing-boxes. Any moisture will, namely, have for effect to absorb the acids formed by the sulphur in the gas, and cause corrosion of the metallic parts.

The Allis-Chalmers Engine.—Lately, large double-acting tandem engines have frequently been installed to work on natural gas, producer gas, or blast-furnace gas. Fig. 158 is a general view of a double-acting twin-tandem four-cycle engine built by the Allis-Chalmers Co. of Milwaukee, Wis.

A cross-section through the cylinder and valve-casings of the Allis-Chalmers engine is shown in Fig. 161, from which the construction of the valve-actuating mechanism may readily be understood.

The valves, it will be seen, are driven from a lay-shaft by means of eccentrics and rolling levers; the latter having for object to effect a quick opening and closing of the valve-ports, though the initial lifting and the seating of the valve is performed only very gradually.

Referring to the set of rolling levers *R* and *F*, which actuate the exhaust valve and which are fulcrumed at *r* and *f*, it will be seen that when the eccentric *E* begins to pull the lever-pin *p* upward, tending to open the valve, the leverage by which the valve is actuated upon is large, at first, and hence the motion of the valve, in starting, slow. As the lever-pin *p* is pulled higher, however, the total leverage decreases gradually, and hence, the speed with which the valve is lifted grows quicker the higher the valve is raised. At the closing of the valve the reversed motion will, of course, obtain. The valve, thus, starting to close quickly, will seat only very gradually. When seated, the valve is held closed by a sufficiently heavy spring and the motion of the rolling levers becomes free and independent of the valve.

The exhaust valve is placed at the bottom side of the cylinder, which is often considered the most suitable arrangement, as it will allow the combustion-chamber to free itself most readily from oil, or impurities that may come with the gas. As generally required in engines above 20 inches, the exhaust valve is water-cooled and its construction is practically identical with that of the water-cooled valve illustrated and described at page 296. The cooling water is supplied by means of a flexible hose connected at *W* and the discharge water is carried off by the same means from the opposite side of the valve-stem.

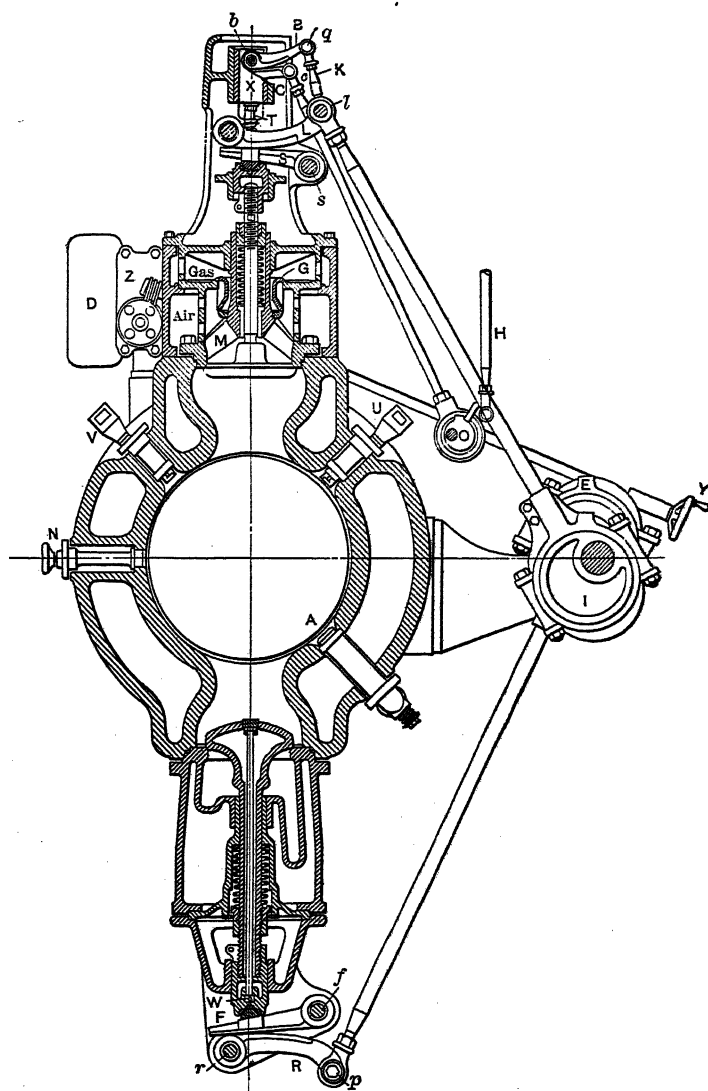



FIG. 161.—Allis-Chalmers Valve Gear.

The gas-valve, *G*, is double-ported and surrounds the main valve-stem. It is actuated from the small crosshead *X* with which it is connected by means of two tie-rods *T*. The gas and air are supplied separately through the manifold *D* to the gas- and air-chambers above and below the gas-valve. The timing of the main inlet valve and gas-valve is such that the main valve opens more or less in advance of the gas-valve at the beginning of the suction-stroke. Thus, at first, air only is admitted to the mixing-chamber and cylinder, and consequently any tendency to back-fire, on account of any lingering flame, or sparks, in the cylinder when the inlet valve opens, will be minimized. After the gas-valve has become opened the mixing of the fuel and air will take place in the mixing-chamber *M;* the gas which is supplied through the gas valve-ports meeting currents of air entering through numerous holes in the walls of the mixing-chamber, in a manner very favorable for effecting an intimate mixture between the two gases. The supply of a suitable fuel-mixture to the cylinder will continue until the gas-valve closes, always a little ahead of the closing of the main inlet valve. The object in delaying the closing of the air-supply being to scavenge the mixing-chamber by pure air from any explosive mixture.

The operation of the valve gear for the inlet valve will be as follows:

The main inlet valve is actuated by means of the inlet eccentric *I* and the rolling levers *L* and *S*, and the motion of the gas-valve will be the same as that of the small crosshead *X* with which it is connected. On the crosshead *X* there is pivoted the rolling lever *B* by means of the pin *b*, while the fulcum lever *C*, on which the former rolls, is forked around the crosshead guide, and pivoted on the valve-bonnet. The lever-pin *q* from which the gas-valve is operated and the lever-pin *l* from which the main valve is operated are connected by means of the link *K*, and, hence, the two levers will be moved in unison by the inlet eccentric.

The fulcrum lever, *S*, for the main valve being solidly hinged on the valve-bonnet, at *s*, the motion of the inlet valve will be uniformly the same for every cycle. The position of the free end *c* of the fulcrum-lever for the gas-valve is, however, under control

of the governor, in such a manner that a small eccentric *O* on the governor-shaft will raise or lower the end of the fulcrum-lever according to the position of the governor. The timing of the opening of the gas-valve, and the lift of the valve, will thus depend on the position of the governor, and will be varied according to the load on the engine. When the governor rises the position of the fulcrum-lever *C* will drop, the opening of the gas-valve will occur later, and its lift will be decreased; and when the governor rises the opposite in each respect will be the occurrence.

The eccentric *O* may be disconnected from the governor-shaft, and swung, by hand, to its lowest position, which will in effect prevent the gas-valve from opening at all. This manipulation becomes of use whenever it may be desired to cut out any one of the combustion-chambers. The rocking motion of the governor-shaft is controlled by means of the connection *H*, whose upper end attaches to the governor-lever.

As is usual with respect to large engines a double set of igniters is provided, located at *U* and *V*. The practice of duplicating the means for igniting the charge has become general, on account of the liability of one set failing, often from a very slight cause.

At *N* is shown a valve-spindle which closes the indicator-opening. By removing this valve-spindle from the outside end of the indicator bushing a thread will be exposed for the attachment of a standard indicator-cock.

At *Z* is shown a valve for the proportioning of the air and the fuel in the charge. The adjustment is made from the operating side of the engine by turning the hand-wheel *Y*. It may become necessary to re-adjust the setting of this valve occasionally during a run, as few fuel-gases in general use are so constant as not to vary quite materially, in heating-value and composition, from time to time.

A check-valve for admitting compressed air for the starting of the engine is provided at *A*. The valve is seated on the inside end of the valve-chamber, and is held closed by a spring acting on the prolonged valve-stem which runs through the chamber. At the outside of the valve-chamber, close to the cylinder, means for the attachment of the compressed-air piping is provided.

The Nuernberg Engine.—A well-known German engine of the same general design as the one just described is the Nuernberg double-acting engine, the latter being, in fact, the original design from which the Allis-Chalmers engine has been developed. A test of the Nuernberg engine on blast-furnace gas is recorded in Table XXXI, pages 410 and 411.

The Westinghouse Engine.—The general construction of the Westinghouse double-acting engine is clearly shown by the half-tone reproduction of a twin-tandem engine, Fig. 162. One of the features of this engine which differs from the Allis-Chalmers-Nuernberg practice is that the outer cylinder-jacket wall is not made continuous from end to end of the cylinder, but is cut circumferentially by the jacket core at the middle of the cylinder, the object being to provide means for the free expansion and contraction of the jacket wall, so as to avoid strains due to the unequal heating of the inner cylinder barrel and the outer wall. The opening in the outer wall is closed by a cast-iron belt which is clamped tightly around the cylinder so as to form the water-jacket.

The governing of the engine is effected by means of a combination throttling and cut-off regulation; the charge of an approximately constant-quality mixture being throttled by the governor to suit the load, and cut off by the closing of the mixing valve.

Referring to Fig. 163, which represents a cross-section through the cylinder and valve casings, it will be seen that the inlet and the exhaust valves are both operated by the same eccentric. The exhaust valve, as is necessary in large engines, is water-cooled; the cooling-water being supplied, and the discharge-water carried off, by flexible hose-connections. The motion of the exhaust valve is transmitted from the eccentric through a pair of rolling levers, *F* and *R*, which give the valve, in lifting it, a rapidly increasing speed, yet starting it from its seat and seating it only very gradually.

The pressure on the valve, when being opened, is approximately 30 to 40 pounds per square inch and, hence, it becomes quite necessary that the first cracking of the valve-opening is performed as gently as possible, to avoid severe strains and jar

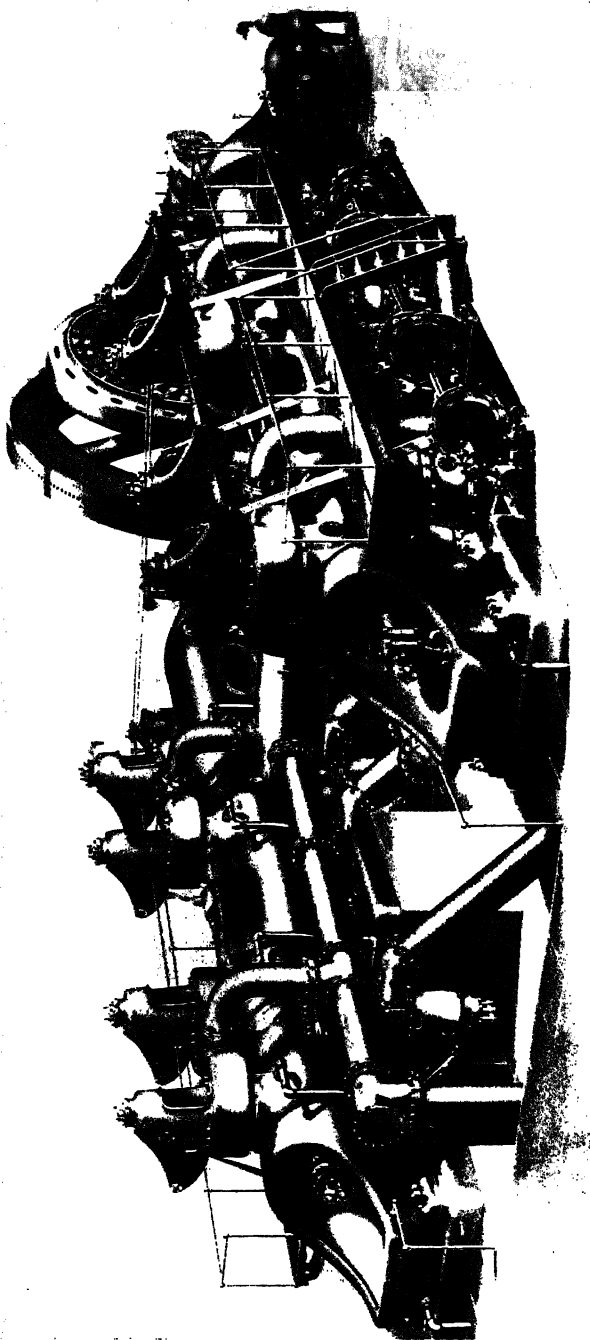



FIG. 162.—Westinghouse Double-Acting Twin-Tandem Engine.

in the valve mechanism. After the valve has become opened, however, there is only the spring-tension to resist its motion and it becomes then desirable to increase the speed with which it is raised, so as to avoid as much as possible the wire-drawing of the discharge.

The admission and mixing valves are actuated by means of the connection *N* and the rolling levers *L* and *K*. The mixing-valve *M* consists, as is seen in the figure, of a cylindrical sleeve mounted on the main admission-valve stem, in such a manner that it follows the motion of the latter up and down, but it is free to rotate about the valve-stem under the influence of the governor. In this sleeve there are provided, for the admission of the gas and air, two sets of rectangular port-openings *o* and *p* which register, when the main valve is fully open and the governor is down, with corresponding ports in the sleeve casing. Between the port-openings of the same set in the sleeve there are bridges somewhat wider than the openings, and if the governor is raised to its highest position the valve-sleeve will be rotated to such a position that when the main valve becomes opened the bridges between the ports in the sleeve will cover the ports in the valve casing and exclude entirely any charge from the cylinder. Between these positions of the mixing-valve, that for the admission of a full charge of air and gas and that for not admitting any charge at all, the governor will have control to rotate it to a position to suit the requirements of the load.

The main admission valve is in the figure shown in its closed position, and it will be noticed that the air-port is covered by a lap somewhat less than that of the gas-port. This will have for effect, that, when the main valve is closing, the gas will be cut off a little earlier than the air, and hence the mixing chamber will, after each suction-stroke, become scavenged by the last incoming air.

The governing of the Westinghouse multiple-cylinder engine of this type, which must be accomplished by revolving four or more mixing-valve sleeves to a position to suit the momentary load, with certainty, is effected indirectly by means of hydraulic power. This becomes necessary, as the valve-sleeves are liable

FIG. 163.—Westinghouse Valve Gear.

to stick, due to collected impurities. The governor proper is of the Hartung type and controls the valve of a small hydraulic cylinder from which the power for adjusting the mixing-valve sleeves is obtained.

The indirect method of governing is described, in detail, at page 314. For an engine of large power the complication involved by this method of governing is of minor importance.

Air-Starting Arrangement.—The starting, with compressed air, of a gas-engine having one combustion-chamber for each stroke of the complete cycle is a very simple matter, as in this case the engine may be started from any position it may have, excepting, of course, from the dead centres. The air-starting arrangement for all multiple-cylinder double-acting engines is in principle the same as that shown in connection with the Westinghouse valve gear, Fig. 163. It consists of a small spring-closing air valve for each combustion-chamber of the engine. This valve is located in the valve-chamber *S*, and its valve-stem, *T*, is prolonged so as to reach close to the small cam, *D*, on the valve-gear shaft, by which it is actuated for opening the valve. The valve-stem is, of course, guided near the cam, at *V*. The timing of the opening and closing of each of the small starting valves is such that air will be admitted to the combustion-chambers during the period of the regular expansion-stroke, when the main admission valve as well as the exhaust valve are closed.

C is a check-valve for admitting to the engine the compressed air for starting. The valve is seated on the inside end of the check-valve chamber, while the outside end of the chamber is connected with an air pipe leading to the starting valve *S*, which is supplied through the opening *A*.

As soon as air is turned on to the starting valves, each one of the combustion-chambers of the engine will, in turn, receive air pressure, and the engine as a whole will work as a compressed-air engine, until the gas-supply valves are opened and the fuel-charge is fired regularly. The compressed air is generally not admitted to the cylinder before the crank has well passed the centre; hence, should the regular fuel-charge explode, usually, in starting, at the time the crank passes the centre, then the check-valve will remain

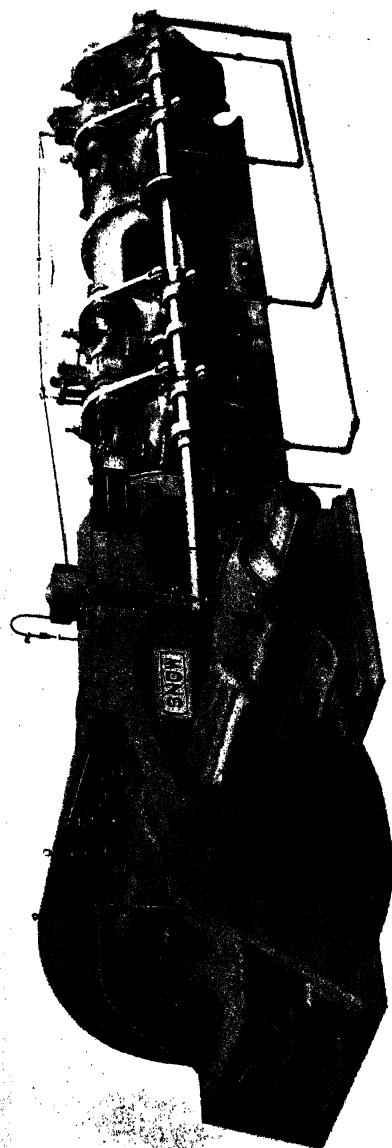



Fig. 164.—"Snow" 500 H. P. Double-Acting Tandem Engine.

closed against the compressed-air admission, due to the explosion-pressure in the cylinder.

The "Snow" Engine.—Fig. 164 illustrates a 22 × 36, 500 horse-power, double-acting tandem four-cycle engine running on illuminating gas, 130 revolutions per minute. It is installed at the People's Gas-Light and Coke Co., Chicago, Ill., and is built by the Snow Steam Pump Co. of Buffalo, N. Y. This engine, it will appear, differs from those of the same type just described in that the valve-chambers are here placed on the side of the cylinder instead of at the top and bottom; an arrangement which has the advantage of leaving the engine foundation continuous from end to end, without any breaks for the accommodation of the exhaust valves.

In Figs. 165 and 166 is illustrated a standard "Snow" valve gear; however, of somewhat different construction from that of the engine shown in Fig. 164. Fig. 165 is a cross-section through the valve-casing and Fig. 166 is a cross-section through the cylinder.

The valve gear is of a release gear type. The inlet and exhaust valves are arranged, the former above the latter, in a valve-chamber, *V*, which is placed at the front side of the cylinder. With respect to the exhaust valve there is no new feature of particular note. It is of a mushroom water-cooled type, and the discharge-water from the valves is carried by a discharge-hose to a jacket for the cooling of the exhaust pipe. A cam-shaft, *A*, running along the front of the engine, drives the valves.

Referring to Fig. 165, the main inlet valve, *I*, and the cut-off or mixing valve, *M*, are arranged side by side in the same valve-casing. The main valve is actuated directly by the valve-rocker, *R*, which is forked around the spring-seat so as to exert a normal force, by pressing on two diametrically opposite points, *a a*, of the lower flange of the spring-seat, when opening the valve. The closing of the valve is effected by a heavy spring, *S*. The upper end of the main valve-stem is connected, by means of a ball-and-socket connection, to one end of the mixing-valve rocker, *U*, the other forked end of which hinges to the crosshead *X*. The rocker, being fulcrumed by means of a pair of links *L*, transmits, thus,

to the crosshead a reciprocating motion in unison with that of the main valve. The crosshead *X* is guided in a bored guide in the upper part of the valve-bonnet, *B*, and it carries the hook arrangement by which the mixing-valve is lifted or released.

The detailed arrangement of the release gear is shown in Fig. 166. On the crosshead, *X*, there is hinged the hook *H*, in such a manner that the catch-block *N* will engage with the block, *O*, on the head of the valve-stem, when the inlet valve is closed. The spring *Z* exerts tension enough on the hook to cause the blocks *N* and *O* to engage, with certainty. When the crosshead is lifted, therefore, the mixing-valve will be lifted with it, until, at the proper point of the stroke determined by the governor, it becomes disengaged. The unhooking is accomplished by the pull-back link, *P*, in connection with a small cam, *C*, on the governor shaft, *G*. The reciprocal action between these two members is readily understood from the figure. The end of the pull-back link will, of course, oscillate with the crosshead and the hook, *H*, to which latter it is hinged; and, from the position in which it is shown in the figure, it can swing only a small angle concentrically with the governor-shaft. If given a greater angular swing the cam-rider, *R*, will ride up on the cam, and carry the link eccentrically in a direction away from the valve-stem head. Therefore, when the crosshead *X* is lifted the valve will follow its motion, until the pull-back link assumes such an angle that the cam-rider is brought far enough up on the cam to cause the hook to be pulled away from the catch-block, *O*, on the valve-stem head. When the governor rises the cam is carried in the direction of the arrow, causing the hook to be pulled out at an earlier point of the stroke; finally so early as never to lift the valve at all. When the hook is pulled, the valve drops to its seat, due to the combined action of gravity and the tension of the spring, *Q*, but, in seating, it will be cushioned by the cushion-piston, *Y*.

While the mixing-valve is open the air and gas pass from the upper and lower valve-chambers and mix in the intermediate mixing-chamber, *D*, and from there the mixture passes through the inlet port to the cylinder. At *E*, Fig. 165, is a hand-wheel for

FIG. 165.—The "Snow" Standard Valve Gear. Longitudinal Section.

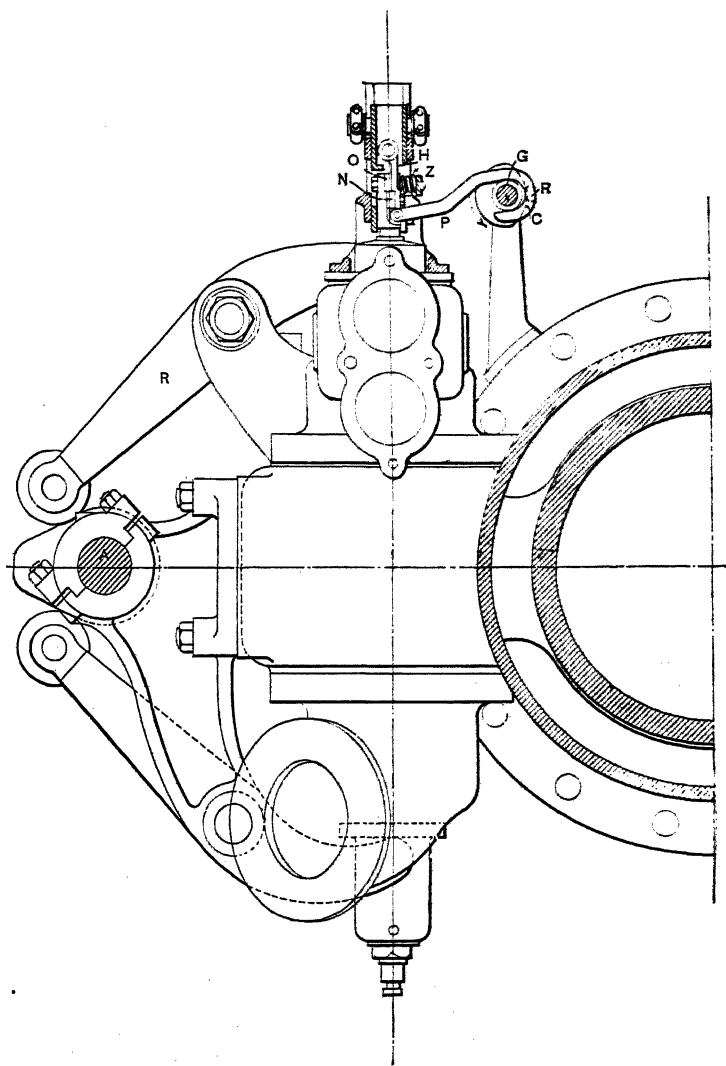



FIG. 166.—The "Snow" Standard Valve Gear. Cross-section through Cylinder.

Reports on Gas-

Name of Installation.	Name of Experimenter.	Locality.	Year.	Cylinder. Dia.	Cylinder. Str.	Rev. per min.	Average M.E.P. Lbs. per sq. in.	Power. I. H.P.	Power. B.H.P	Mech. Eff'y.
Illuminating Gas—English Engines.										
National	Robinson	Ashton	1898	10	18	170	87	26.4	23	87
Tangye	Witz	Lille	1902	10	19	193.6	79	28.6	24.5	85
Crossley	Atkinson	Openshaw	1894	11.5	21.0	173	98	46.5	39.9	86
Crossley	Clerk	Openshaw	1894	7	15.0	200	96	14	12	86
Stockport	Bellamy	Reddish	1895	9.4	17.0	184	90	24.5	20.8	85
Illuminating Gas—French and German Engines.										
Deutz, Otto	Meyer	G. M. Fabr. Deutz	1898	15	22.8	199	73	73 2	65.1	89
Koerting	Meyer	Hanover	1899	6.8	13.3	221	78	11.7	10.4	89
Niel	Witz & Moreau	Evreux	1901	13.7	18.9	213	71	53.5	46	86
Gyldner	Schroeter		1904	10	15.7	210.7	106	35.9	......	
Natural Gas.										
Westinghouse	Millar & Gladden	Pittsburg	1899	25	30	147	61	677	606	90
Westinghouse	Robertson	Lafayette	1900	13	14	265	60	113	92	79.4
Snow Pump W.	Hastings & Parker	Halsey	1901	25	48	....	......	736	595	80.7
Coke-Oven Gas.										
Borsig-Oechelhaeuser	Meyer	Silesia	1905	26.6	37.5	110.6	75	Total 878 net 765	628	Total 71.5 net 82.1
Producer Gas.										
Mond Gas Premier Eng.	Humphrey	Winnington	1900	28.$\frac{1}{8}$	30	128	81	480	368	Total 76 net 81.2
Koerting Gen. Koerting Eng.	Meyer	Hanover	1900	21.6	37.7	101	68	481	341	71
Dawson Gen. Grossley Eng.	Meyer	Zurich	1895	16.0	24	165	58	64	58	90
Westinghouse Eng. Loomis-Pettibone Gen.	Alden & Bibbins	Worcester	1907	23.5	33	150	55	600	500	83.5
Blast-Furnace Gas.										
Cockerill	Hubert	Seraing	1900	51	55	93	56	886	725	82
Deutsche Kraft A. Gesellschaft	Meyer	Differdingen	1898	17	27.5	160	63	79.5	67.6	85
Nuernberg	Riedler	Rombach	1904	33.5	43.4	105.6	70	1427	1186	83.1
Oil Engines.										
Banki	Taborsky	Buda Pest	1901	9.8	15.7	209	......	......	26.38	.....
Diesel	Ade Clark	Ghent	1903	.....	.....	154	......	203	164	80.7
Diesel	C. Eberly	Augsburg	1908	.....	.....	402.4	......	392	297	76.2
Diesel	Meyer	....	1903	15.75	23.65	158.8	95	88	69.6	79.1
Alcohol Engines.										
Deutz	Meyer	....	1903	8.35	11.8	276.9	......	16.8	......	......
Marienfelde	Meyer	....	1903	9.95	15.75	197.6	......	19.77	......	......
Banki	Schimanek	Buda Pest	1903	.....	.....	225	......	......	32.13	......

Engine Performances.

Heating Value of Gas B. T. U. per c. f. or Lbs.	Fuel Used Per Hour.		Heating Value Used Per Minute.		Thermal Efficiency B.T.U.		Engine Cycle.	Type. Single or Double Acting.	Reference.
	per I.H.P. c. f.	per B.H.P. c. f.	per I.H.P. B.T.U.	per B.H.P. B.T.U.	per I.H.P.	per B.H.P.			
630	14.0	16.1	148	170	28.8	25.0	4	*S*	Rep. Pr. Robinson.
609.7	14.5	17	147.3	172.7	28.8	24.5	4	*S*	Aimé Witz.
640	14.5	16.5	155	176	28	24	4	*S*	Eng'g. Nov. 30, 1894.
680	14.8	17	168	193	25.6	22	4	*S*	P. I. C. E. Vol. 124.
642	19.0	22.3	203	238	21.2	18	4	*S*	P. I. C. E. Vol. 124.
557.5	13.7	15.4	127.3	143	33.3	29.6	4	*S*	Z. V. D. I. Vol. 43.
496.5	16.0	18.0	132.4	149	32.0	28.4	4	*S*	Z. V. D. I. Vol. 43.
636.6	13.3	15.5	141	164.5	30.3	26	4	*S*	Aimé Witz.
....	......	......	......	......	42.7	......	......	......	Z. D. V. I. Vol. 48.
H. H. V. 1000	8.9	10	148.3	166.6	28.6	25.5	4	2 Cyl. *D*	Sibley Journal, 1900.
H. H. V. 970	10.62	13.69	171.6	221.2	24.7	19.3	4	3 Cyl. *S*	Bryan Donkin and Trans. A. S. M. E. 1900
H. H. V. 1175	7.37	9.13	144.3	178.8	29.4	23.7	4	4 Cyl. *S*	Sibley Journal, 1901.
380	17.8	25	Per T. 112 Per N 128	157	Total. 37.9 net. 33	27.5	2	1 Cyl. *D*	Z. V. D. I. Vol. 49.
144	52	69	125	166	33.7	25.6	4	2 Cyl. S. T.	Proc. I. M. E. 1901.
129	57.9	81.5	125	176	34	24.1	2	*D*	Gas L. Jour. 1900.
144	1.22 lbs.	1.36 lbs.	......	......	24	21.5	4	*S*	Z. V. D. I. Vol. 39.
106	79	95	140	168	30.1	25.2	4	2 Cyl. *D*	Trans. A. S. M. E. 1907.
97	83	101	134	163	31.5	26.0	4	*D*	Bul. de la Soc. Ind. Min. Vol. 14.
105	79	63	139	164	30	25	4	*S*	Z. V. D. I. Vol. 43.
88	85	102	125	150	33.9	28.2	4	2 Cyl. *DT*..	Gross-Gas-Maschinen, Riedler.
Benzine 18300 per lbs.	......	0.48 lbs.	......	148	......	28.5	4	*S*	The Engineer, 1903.
Crude oil 19300 per lbs.	0.33 lbs.	0.41 lbs.	107	131	40	32.3	....	......	Proc. I. M. E.
Kerosene. 18130 per lb.	0.302 lbs.	0.43 lbs.	91.2	130	45.8	32.2	4	4 Cyl. *S* Cr. 180°	Z. V. D. I. Vol. 52.
Kerosene 18610 per lb.	......	......	......	......	40	31.9	4	*S*	Z. V. D. I. Vol. 47.
9900 per lb.	......	0.813 lbs.	......	.....	....	31.6	4	*S*	Z. V. D. I. Vol. 47.
9900 per lb.	......	0.786 lbs.	......	.....	....	32.7	4	*S*	Z. V. D. I. Vol. 47.
9700 per lb.	......	0.87 lbs.	......	.....	.....	30.2	4	*S*	Z. V. D. I. Val. 47.

the adjustment of the throttle-valve, T, in the gas-port, by which the proportioning of the gas and air is effected.

Operating as described, this regulation effects a constant-quality mixture, but, if desirable, in order to obtain a more constant compression, the mixture may be diluted at light loads, by arranging, in the gas-supply pipe, a butterfly valve that will be controlled by the governor.

The Cockerill Engine.—Another well-known double-acting, four-cycle engine of the same general type as the ones described in the previous is the Cockerill engine, built by the Cockerill Co. of Seraing, Belgium. Similarly to the Nuernberg engine, and to German engines in general, the Cockerill construction employs a main centre-crank, which allows the shaft, as well as the main engine frame, to be built lighter than would be required for an overhung crank. The construction involves, however, difficulties in the maintaining of the shaft-journals in proper alignment, that must be taken into consideration when judging the relative advantages between the American and European practice in this respect.

In Table XXXI are recorded some performances of gas-engines of various types and on different fuels. The highest efficiencies recorded for each group are, approximately, the best that has been obtained for the particular fuel, whereas the low figures, in the majority of cases, represent efficiencies that may be expected under ordinary good conditions, and at full load.

The average M.E.P. has been computed from the power generated and the total volume, per minute, of the pressure strokes. Generally, the tests recorded have been made under approximately full-load conditions; therefore, the M.E.P. is, in most cases, the maximum. In some cases, however, the M.E.P. is very low, showing that the average load has been below the maximum, or a poor mixture has been used.

The alcohol engines recorded in the table are all of high efficiency, due to a very high compression of the charge, that has been made possible by injecting water in to the cylinder with the fuel-charge.

The compression ratios employed for the three alcohol engines recorded are:

The Deutz engine 1 to 8.9.

The Marienfelde engine 1 to 10.26.

The Banki engine 1 to 10.

Denatured alcohol was used in all three tests; in the two first the hydration was 91.2 per cent alcohol, by volume, and in the third it was 87 per cent alcohol, by volume.

CHAPTER XV

PRODUCER-GAS AND GAS-PRODUCERS

Introductory.—Producer-gas was used, as early as 1857, by Siemens as fuel in his steel-furnaces, and in connection with his regenerative furnace it was found to be a most convenient and economical means for obtaining the high temperatures which he required in his steel-making process.

Although economical as fuel, the Siemens gas is not efficient for use in the gas-engine. For the reason that, when used as fuel, not only the potential heating-value of the gas, but also its sensible heat, becomes available heating-value; whereas, as the gas can be used in the gas-engine only after having been cooled, a great percentage of its total heating-value is thrown away in the cooling-process.

The year 1881 can be counted the beginning of the era of producer-gas power. Mr. Dawson exhibited that year his first successful power-gas producer, in connection with a small Otto gas-engine, and the success of the exhibit was such that producer-gas came soon afterward to be considered a most efficient competitor with steam as a means for generating power from available fuels.

Since then, improvements have been made in the gas-producer as well as in the gas-engine, until, to-day, producer-gas is extensively used with success, in many instances on a large scale in preference to steam.

The principal difference between the way in which Siemens generated his fuel-gas and the way in which power-gas is obtained is that in the modern gas-producer steam is introduced into the furnace, for the purpose of cooling the fuel-bed to some extent, and thus prevent an excessive loss of heat in sensible heat of the gases, that will be wasted in the subsequent cooling-process.

The Gas-Producer.—The simple suction gas-producer, shown

in Fig. 167, as well as, in a general way, the pressure producer, may be described as an air-tight steel-plate cylinder containing at the bottom an ash-pit, a grate for supporting the fuel—sometimes a shaking-grate—and, above, a substantial firebrick-lined furnace. Above the furnace there is a fuel-hopper, and a charging bell seals the opening through which fuel is charged into the producer. There are also poke-openings provided on top of the producer-furnace, through which poke-bars may be inserted for the purpose of poking down or breaking up the fuel-bed, which may be held up or made solid by the caking or clinkering of the fuel. At a suitable height above the grate, there is a gas-outlet taken for conveying the generated gases from the producer in to the necessary cooling and cleaning apparatus.

When in normal operation, there is on the grate of the furnace a thin bed of ashes and, above, a deep incandescent fuel-bed on top of which a layer of fuel that has not as yet attained high incandescence should preferably be in evidence.

The only access for air for maintaining combustion in the producer is generally through the ash-pit, and the draft through the fuel-bed must be regulated to that necessary for generating the required amount of gas of a proper quality. The draft is maintained either by the suction of the gas-engine piston, by a pressure-blower, or by an exhauster of some kind.

The Process.—The result of this arrangement of blowing air in to, or drawing a current of air through, a deep bed of incandescent fuel will be as follows:

Near the grate-level there is formed, through ordinary combustion, carbon dioxide; a gas often referred to as $C\,O_2$-gas, and which, as its name implies, is composed in such a manner that each molecule, or element of the gas, consists of one atom of carbon and two atoms of oxygen.

In being drawn through a deep, porous bed of highly heated carbon, this gas absorbs readily more carbon, so that new molecules, consisting each of one atom of carbon and one atom of oxygen, are formed. This new gas, carbon monoxide or $C\,O$-gas, which is always the result of incomplete combustion of carbon, is the main combustible in producer-gas.

By incomplete combustion of one pound of carbon there is formed 2⅓ pounds of carbon monoxide gas, and the heat generated is about 4,380 heat units. The carbon monoxide gas formed from each pound of carbon, at the primary combustion in the producer, requires for its combustion the same amount of oxygen as that consumed by the original pound of carbon in its gasification to CO, or 1⅓ pound, and the heat generated by the combustion is about 10,200 heat units. It will, thus, be seen that the main combustible of producer-gas contains a potential heating-value of about two-thirds of the heating-value of the fuel, if the latter, as has been assumed, consists of pure carbon only.

The carbon monoxide is, however, not all the heating-value obtained in the gas at the primary gasification, because the hydrocarbons which most fuels contain, and which are distilled off into the gas, enriches it, and, further, the sensible heat of the gas adds materially to its total heating-value.

At the primary combustion there is liberated, as stated, about 4,380 heat units for each pound of carbon gasified to CO-gas. This heat will, of course, tend to heat up the fuel-bed and the producer, until the radiation from the apparatus, together with the sensible heat that is carried off by the gases, will just balance the heat liberated.

The Siemens producer for fuel-gas was operated according to the preceding outline, and it generated a gas low in potential heat-value but of a high temperature. That is, the gas carried off, when leaving the producer, a considerable amount of heat as sensible heat.

In the modern producer the object is to generate combustible gas of as high heating-value as possible, when cold, but any sensible heat is not required for the process, excepting to the amount necessary for maintaining the temperature of the furnace at such degree that the formation of CO-gas takes place readily. For the formation of combustible gas the producer must be supplied with the amount of air, only, that is required for the gasification of carbon to carbon monoxide gas. But only part of the heat generated during this gasification, necessarily only about

1,900* heat-units, will be carried away by the gases, or dissipated through radiation, when the temperature of the furnace is such as is normally required for efficient gasification. There will, therefore, be a surplus of heat, about 2,500 heat-units per pound of coal consumed, that must be abstracted from the fuel-bed, in order not to overheat the same as the process of gasification proceeds. This surplus heat can be utilized in a most efficient and desirable manner, simply by the introduction of steam in to the furnace.

Steam, being subject to decomposition into its constituent elements, oxygen and hydrogen, when in contact with incandescent carbon of sufficiently high temperature, is a most useful agent in a gas-producer. It serves, primarily, three good purposes; in that it absorbs the surplus heat from the furnace at its decomposition, in that it enriches the gas with its hydrogen, and in that it furnishes oxygen for the gasification of carbon.

The oxygen which is furnished to the gas-generator by the introduction of air carries with it nitrogen in the proportion of 3·33 pounds of nitrogen for each pound of oxygen, and this nitrogen, which dilutes the gas, will be of no value as far as the heating-value of the gas is concerned. It is, therefore, evident that the more oxygen that can be obtained for the combustion of carbon by decomposition of steam the richer in heating-value the generated gas will be.

There are, however, only about 2,500 heat-units available for the decomposition of steam and for the formation of hydrogen, and, as 327 heat-units are required for the formation of each cubic foot of hydrogen, there is a limit of 8 cubic feet of hydrogen that can be obtained per pound of carbon by utilizing the surplus heat only for its formation.

As from each pound of carbon there will be generated $2\frac{1}{3}$ pounds, or 32 cubic feet of carbon monoxide, the volume of hydrogen will, therefore, be about one-fourth of that of the carbon monoxide, when all the fuel is burned to this gas.

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*This figure is approximate only; its value having been estimated with reference to the average conditions obtaining at an efficient gas-making process as carried out at present. See page 423.

Generally we find, however, producer-gas in which the volume of hydrogen is more than one-fourth of that of the carbon monoxide, and this may be taken as evidence that some of the fuel has been subjected to complete combustion to carbon dioxide-gas. When the volume of the carbon dioxide in producer-gas is not over five per cent, then there has been no perceptible loss incurred by having burned the carbon completely to carbon dioxide, because the additional heat generated thereby has been utilized in forming a corresponding additional percentage of hydrogen, which enriches the gas.

What we call producer-gas consists, as will appear from what has been stated, principally of five separate elementary products, each of a very distinct nature.

These are:

Carbon monoxide, which is obtained at the primary gasification of carbon;

Hydrogen, which has been formed by reclaiming the heat liberated at the primary gasification by decomposition of steam;

Hydrocarbons, which have been distilled off from the fuel;

Nitrogen, obtained from the air consumed, and

Carbon dioxide gas formed at the complete combustion of some of the fuel.

Of these elementary products, the three first are combustible gases, and they compose somewhat more than 40 per cent of the total volume of the producer-gas.

Nitrogen and carbon dioxide are inert gases and compose nearly 60 per cent of the total volume of the generated gas.

Producer-gas has in some rare cases been found to contain, when hot, as much as 93 per cent of the heating-value of the fuel from which it has been formed, but, when cooled and ready for use in the gas-engine, it carries rarely more than 85 per cent of the heating-value of the fuel.

The sensible heat of the gas when it leaves the producer is, in part, generally utilized for heating the fresh fuel charge, for pre-heating the air and for vaporizing the steam that is supplied to the gas generator.

Heat-Transfer of a Theoretical Gas-Making Process.—The

waste of fuel in the ashes, and the waste of heat through radiation are the only direct losses of heating-value in the gasification of fuel. In the process itself there is no loss or gain, only a transfer of heating-value, from the carbon, in to the gases generated. Theoretically, the gasification may be made, with equal economy, either into carbon monoxide and hydrogen or into carbon dioxide and hydrogen, and that this is so will readily be seen by the following representations of the heat-transfers that must be due to the reactions taking place in the two cases.

It is assumed, in the first case, that 12 pounds of carbon are gasified by the oxygen derived through dissociation of 18 pounds of water into 2 pounds of hydrogen and 28 pounds of *C O*-gas.

The total potential heating-value of 12 pounds of carbon is $12 \times 14,600 = 175,200$ B.T.U., of which:
there are evolved, at the combustion of 12 pounds of carbon to *C O*-gas, $12 \times 4,380 = 52,560$ B.T.U.;
the balance, 122,640 B.T.U., is transferred to the *C O*-gas.

The dissociation of 18 pounds of water requires $18 \times 6900 = 124,200$ B.T.U., and of this amount 52,560 B.T.U. become available at the combustion of the carbon, the rest, 71,640 B.T.U., must be supplied from outside source. The total amount, 124,200 B.T.U., absorbed at the dissociation of the water will be found as potential heating-value in the hydrogen formed.

The figures above are based on the high calorific value of hydrogen, and include therefore the heat required for the vaporization of the water into steam.

When carbon is gasified into *C O*-gas and hydrogen the following theoretical heat-transfer will take place:

12 lbs. C +	18 lbs. H_2O =	2 lbs. H_2 +	28 lbs. CO
B.T.U.	B.T.U.	B.T.U.	B.T.U.
Potential value. 175,200	Absorb at dissociation:		
Evolved at comb. to CO 52,560	From outside source..... 71,640		
Transferred to CO-gas 122,640	From comb. of C....... 52,560	Potential value. 124,200	Potential value. 122,640
	Transferred to H_2 124,200		

Hence $\frac{52,560}{124,200}$ (= 0.42) of 18 pounds of water is dissociated by heat evolved at the combustion of 12 pounds of carbon to CO-gas, and 58 per cent must be decomposed by outside heat-supply, or the heat evolved from each pound of carbon will dissociate 0.63 pound of water.

It is evident, therefore, that, at a theoretical gasification, 42 per cent of the carbon will be gasified by oxygen liberated from water due to the heat evolved from the carbon itself, and 58 per cent must be gasified by oxygen derived from the atmosphere.

If, secondly, the carbon is gasified to CO_2-gas by oxygen derived from water, then 12 pounds of carbon will require the dissociation of 36 pounds of water to form 4 pounds of hydrogen and 44 pounds of CO_2-gas.

In the gasification of carbon into carbon dioxide and hydrogen, the theoretical heat-transfer will be as follows:

12 lbs. C	B.T.U.	+	36 lbs. $2H_2O$	B.T.U.	4 lbs. $2H_2$ B.T.U.	+	44 lbs. CO_2 B.T.U.
Potential value	175,200		Absorb at dissociation:				
Evolved at comb. to CO_2	175,200		From outside source.....	73,200			
Transferred to CO_2-gas..	0		From comb. of C.	175,200			
			Transferred to H_2.	248,400	Potential value. 248,400		Potential value. 0

Hence $\frac{175,200}{248,400}$ (= 0.70) of 36 pounds of water is dissociated by heat evolved at the combustion of carbon to CO_2-gas and 30 per cent must be decomposed by outside heat-supply; or by the heat evolved from each pound of carbon there will be dissociated 2.1 pound of water.

Thus, 70 per cent of the carbon will be gasified by oxygen liberated from water due to the heat evolved from the carbon itself, and 30 per cent must be gasified by oxygen derived from the atmosphere.

Composition of Gas Resulting from Gasification without Heat

Loss.—A theoretical gasification of one pound of C to H, CO and N will result in the following composition:

	Pounds.	Per Cent. Weight.	Cubic Feet.	Per Cent. Volume.
CO	2.333	0.47	31.64	0.40
H	0.07	0.015	13.30	0.17
N	2.57	0.515	34.85	0.43
	4.97	1.00	79.79	1.00

Heating-value per cubic foot 184 B.T.U.

The theoretical composition of a gas resulting from the gasification of one pound of C to CO_2, H and N will be the following:

	Pounds.	Per Cent. Weight.	Cubic Feet.	Per Cent. Volume.
CO_2	3.67	0.56	31.64	0.28
H	0.234	0.036	44.27	0.40
N	2.66	0.404	36.07	0.32
	6.56	1.000	111.98	1.00

Heating-value per cubic foot 130 B.T.U.

An examination of the weights of the elementary gases obtained in each case will show, that the heating-value of the total resulting composition is in both cases the same; the heating-value of 2.333 pound of CO being the same as that of 0.164 pound of H.

The weight of the resulting gases is, however, in the latter case considerably more than in the former.

During the practical process of generating producer-gas there is always some loss, due to heat-radiation and due to cooling of the gas, and the greater the weight of the final product the greater will be the heat-loss incurred. It is evident, therefore, that the process of generating H and CO_2-gas will involve a greater loss than the process of generating H and CO-gas.

In the following tables are given the composition and heating-value of producer-gas generated from carbon, with varying percentages of the fuel burned to CO_2-gas. The assumption has been made that in producing CO-gas an efficiency of 85 per

cent is obtained, and that in producing CO_2-gas and H the efficiency will be 70 per cent.

Composition of Producer-Gas Containing Varying Percentages of CO_2.—Increasing percentages of the fuel are assumed to have been burned to CO_2-gas, with correspondingly diminished efficiency.

Per Cent C Gasified to		Products Derived from 1 lb. C, in Pounds.				Total Weight of Products from 1 lb. C, in lbs.	Heating Value per pound of Gas.
C O.		C O.	H.	N.	CO_2.		
100		2.333	0.035	3.5	0	5.868	2120
95		2.216	0.042	3.55	0.183	5.991	2053
90	10	2.099	0.048	3.6	0.367	6.114	1990
85	15	1.982	0.055	3.65	0.55	6.237	1935
80	20	1.865	0.061	3.7	0.733	6.359	1882
75	25	1.748	0.068	3.75	0.917	6.483	1830

Per Cent C Gasified to		Products. Percentage Volume.				Volume of Products from 1 lb. C, in cub. ft.	Heating Value per cub. ft. B.T.U.
C O.	CO_2.	C O.	H.	N.	CO_2.		
100	0	37.0	7.7	55.3	0	85.7	145
95	5	43.3	9.0	54.9	1.8	87.7	140
90	10	31.8	10.2	54.5	3.5	89.6	136
85	15	29.4	11.3	54.1	5.2	91.5	132
80	20	27.1	12.4	53.7	6.8	93.4	128
75	25	24.9	13.4	53.4	8.3	95.3	125

Quantity of Steam to be Supplied.—Eighty-five per cent is on an average the best return in heating-value obtained in a cold gas generated from an anthracite fuel of, say, 14,000 B.T.U. The heat-loss per pound of fuel is, thus, 2,100 B.T.U.

The principal wastes are: (1) in the sensible heat carried off by the gases; (2) in loss of fuel in the ashes, and (3) in heat dissipated through radiation.

An approximation of the amount of the first item is readily obtained, as it is known that, normally, 5½ pounds of gases are

produced per pound of carbon, and that they leave the producer, normally, at about 850° F. Their specific heat is, on an average, 0.25. The air and vapor being supplied at a temperature of, at least, 150 degrees, the range of temperature through which the gases are heated in the furnace is about 700 degrees.

The heat-loss, therefore, due to item (1), is 962 B.T.U. per pound of fuel. The waste of fuel should not exceed 1½ per cent, or 210 heat-units per pound of fuel. The total loss due to items (1) and (2), thus, 1,172 B.T.U. If this heat-loss be subtracted from the total loss 2,100 B.T.U. there remains 928 B.T.U., which is the loss due to radiation. Hence, it may be said that approximately 1,900 B.T.U. are wasted in the cooling of the gases and through radiation. The approximate value of this item of heat-loss is of interest, as it has bearing on the quantity of steam that should, normally, be supplied to the furnace.

Deducting 1,900 B.T.U. from the heat evolved at the gasification of the fuel to CO, 4,380 B.T.U., the remaining 2,480 B.T.U. becomes available for decomposition of steam.

It has been found, however, that for the reduction of all the carbon dioxide formed in the lower part of the furnace it is necessary that the temperature of the fuel-bed be very high, and the current through it slow. A high furnace temperature will, on the other hand, result in increased loss due to the higher temperature at which the gases are carried off and in increased radiation losses. Besides, difficulties will result from clinkering of the refuse in the fuel.

Practice has shown that an average temperature of the fuel-bed of 1,800 to 2,000 F°*. will give best results, but, at such low furnace temperature, there will remain in the gas an appreciable amount of CO_2-gas, unreduced.

In an average, favorable case seven per cent of the carbon will be burned to CO_2-gas and hence an additional heating-value of 715 B.T.U. per pound of fuel will be available for decomposition of steam. The total available heating-value, thus, 3,195 B.T.U.

* Compare: "Producer Gas," by J. Emerson Dawson, pp. 6 and 45.

The weight of the steam to be supplied is, therefore, $\frac{3,195}{6,900}$ (= 0.463) pounds, per pound of fuel. At the decomposition of this steam there is formed $\frac{8}{9} \times 0.463$ (= 0.412) pound oxygen, which will effect a gasification of $\frac{0.412}{1.33}$ (= 0.31) pound of carbon.

Accordingly, there will be required normally not more than 0.46 pound of steam per pound of carbon, and the oxygen derived at its decomposition will gasify, on an average, 0.31 pound of carbon.

The specific heat of steam being high, a material loss will be incurred by supplying an excess amount of steam, above what can be decomposed by the fuel. The excess will only, in passing through the fuel-bed, absorb heat as superheat, which will be wasted at the subsequent cooling of the gases. Besides, it will cool the lower part of the fire below the required temperature for efficient reduction of the CO_2-gas, and after the fire has been driven to the top of the fuel-bed due to excessive steam-supply, there will generally be experienced some little difficulty to get the fuel-bed in condition again for producing a proper gas.

Exhaust-Gas from the Engine used in the Producer Instead of Steam.—Instead of using steam as a means for reclaiming the heat generated at the primary combustion in the producer, the exhaust gases from the gas-engine have been successfully employed for the purpose. The process consists in the reduction of the CO_2-gas, returned to the producer from the engine-exhaust, into CO-gas. The gas generated by this method, consisting principally of CO-gas and nitrogen, will stand a high compression, but, being very lean in heating-value, it will require somewhat larger cylinder capacity than ordinary producer-gas. On this account, it is doubtful if the increased cost of the installation does not outweigh the gain due to any increased efficiency that may be derived through the employment of a high compression.

The saving due to the utilization of the sensible heat of the exhaust gases cannot be material, particularly as it is seldom

convenient to locate the engine close enough to the producer to fully take advantage of this item; moreover, the gasification process requires the return to the furnace of only a part of the total volume of the exhaust, and in it is included a considerable amount of nitrogen that becomes reheated in the furnace, and generally leaves the producer at a higher temperature than that at which the exhaust gases are admitted.

It can readily be ascertained that, theoretically, the gas generated by the method of by-passing the exhaust from the engine in to the furnace can never be richer in heating-value than gas obtained through gasification with air only, according to Siemens process. The heat of the primary gasification can be reclaimed, but the heating-value, per cubic foot of gas, will not be enriched thereby. A demonstration, as the following, will show this.

For each cubic foot of CO-gas generated by the combustion of carbon, figured at 62° F., there will be evolved 140 B.T.U. Assuming all this heat to be utilized for the decomposition of CO_2 of the exhaust, then, as there is required 324 B.T.U. for the decomposition of one cubic foot of CO_2 into one cubic foot of CO and ½ cubic foot of O, there will be obtained

$$\frac{140}{324}(1\ c.f.\,CO + \tfrac{1}{2}\ c.f.\,O) = 0.43\ c.f.\ CO + 0.215\ c.f.\ O.$$

Each cubic foot of CO-gas formed by combustion of C with air will carry 1.88 cubic foot of nitrogen, whereas each cubic foot of CO reduced from CO_2-gas will carry twice that amount, or 3.76 cubic feet. This is evident, because the CO_2 in the exhaust has consumed twice as much air as that required for the combustion of carbon to CO-gas. The process that will take place at the gas-production will be the following.

In the production of 1 cubic foot of CO-gas from carbon there will be generated:

0.57 *c.f.* of CO by air, carrying $0.57 \times 1.88\ N =$	$1.07\ N.$
0.43 *c.f.* of CO by O, carrying	*no* $N.$
0.43 *c.f.* of CO reduced from CO_2, carrying $0.43 \times 2 \times 1.88\ N =$	$1.62\ N.$
1.43 *c.f.* of CO in total, carrying	$2.69\ N.$

Hence, each cubic foot of CO will carry 1.88 cubic foot of N, or, in other words, the heating-value per cubic foot of the products will be at par with that of air-gas.

The above computation may appear complicated, but it can be simplified if the following facts are recognized:

That 1 cubic foot CO-gas obtained by the gasification of C with air carries 1.88 cubic foot N.
That 1 cubic foot CO_2-gas of the exhaust will carry at least 2×1.88 cubic foot N;
That ½ cubic foot O will give with C 1 cubic foot CO.

Assuming, then, that, through combustion of carbon with air and with O, there has been evolved, in the furnace, heat enough for the decomposition of just one cubic foot of CO_2, thus,

1 cubic foot of CO_2 becomes decomposed to
1 cubic foot of CO, carrying 2×1.88 N,
and ½ cubic foot of O, giving with C
1 cubic foot of CO, carrying *no* N.

Hence, the advantage gained in obtaining free oxygen through dissociation of CO_2-gas, is exactly balanced by the increased amount of nitrogen which the CO_2-gas carries in to the furnace.

It will be evident that for reclaiming all the heat of the primary combustion there will be required $\left(\frac{43}{143} = \right)$ one-third of the total volume of the exhaust from the engine.

Ninety cubic feet of Siemens air-gas generated from one pound of carbon contains 10,200 B.T.U., or the gas is of 113 B.T.U. per cubic foot.

There is added in ordinary producer-gas, theoretically, 8 cubic feet of hydrogen of a heating-value of, in all, 2,500 B.T.U., making the gas of a heating-value of 130 B.T.U. per cubic foot, or 15 per cent richer than air-gas. Practically, due to partial combustion of the fuel to CO_2, by which an additional volume

of hydrogen is formed, the ordinary producer-gas becomes richer still.

On the other hand, when the difficulty of regulating the volume of the exhaust returned to the furnace to the correct proportions is considered, it may be expected that, practically, the gas produced when the engine exhaust is returned to the generator would become even of a comparatively poorer quality than the theoretical computation tends to show. But the fact is that the gas, actually, will be enriched by an appreciable amount of hydrogen derived from the moisture in the fuel and in the air charged to the furnace, as well as from the moisture in the air charged to the engine, which will appear in the exhaust gases.

The best efficiency so far reported to have been obtained by this gas is about at par with that obtained by ordinary producer-gas.*

The Composition and Heating-Value of Producer-Gas.— Theoretical Analysis of Anthracite Gas.—The normal composition of producer-gas may be determined, approximately, from the analysis of the fuel from which it is made.

With respect to the hydrocarbon contents, the composition of anthracite gas will be approximated by assuming that all the hydrocarbons are distilled off from the fuel, and that they will be found in the gas in the form of marsh-gas.

Assume the analysis of an anthracite coal to be:

Fixed carbon	84 per cent
Volatile hydrocarbons	4 per cent
Ash and moisture	12 per cent

It is safe to assume that 25 per cent of the fixed carbon will be gasified to CO-gas by oxygen derived from steam, 65 per cent gasified to CO-gas by oxygen derived from the atmosphere, and that 10 per cent is burned to CO_2-gas.

* Compare the efficiency reported by Mr. G. M. S. Tait, page 795, Tr. n. Soc. Mech. Eng., Vol. XXX, with producer-gas reports, Table XXXI.

	Pounds of Gas.	Per Cent. Weight.	Cub. Feet Gas at 62° F.	Per Cent. Volume.
Hence, of 100 pounds of fuel:				
21 lbs. C gasified to CO by steam require 28 lbs. O, giving CO	49.0			
54.6 lbs. C gasified to CO with air require 72.8 lbs. O, giving CO	127.4			
TOTAL CO.......	176.4	33.22	2392.0	31.04
8.4 lbs. C burned to CO_2 require 22.4 lbs. O, giving CO_2.....	30.8	5.80	265.8	3.45
4. lbs. hydrocarbons distilled give CH_4....................	4.0	0.75	94.9	1.23
88 lbs. combustible.				
28 lbs. O dissociated from steam liberate H..................	3.5	0.66	665.0	8.63
95 lbs. O derived from the atmosphere carry N............	316.3	59.57	4289.0	55.65
	531.0	100	7706.7	100

The heat-energy in the gas will be

in 176.4 lbs. CO at 4,380 B.T.U. per pound	772,632	B.T.U.
in 4 lbs. CH_4 at 21,900 B.T.U. per pound	87,600	B.T.U.
in 3.5 lbs. H at 62,100 B.T.U. per pound	217,350	B.T.U.
Total	1,077,582	B.T.U.

Heat-energy per pound gas	2,030 B.T.U.
Heat-energy per cubic foot at 62° F.	140 B.T.U.
Heat-energy in the gas from one pound of coal	10,775 B.T.U.
Heat-energy in one pound of coal	13,140 B.T.U.
The efficiency of conversion into gas	0.82

The following may be considered as an average composition of anthracite suction gas:

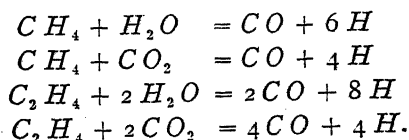
	Per Cent, Weight.	Per Cent, Volume
CO	27.6	26.0
H	0.7	9.0
CH_4	1.2	2.0
N	60.5	57.0
CO_2	10.0	6.0
	100.0	100.0

Heating-value per cub. ft. at 62° F., and 1 atmosphere 134 B.T.U.
Volume per pound of gas at 62° F., and 1 atmosphere 14.5 cub. ft.

Theoretical Analysis of Bituminous Gas.—With regard to bituminous coals containing a high percentage of volatile hydrocarbons of various compositions, there exists an uncertainty as to what amount of the latter will become oxidized during the combustion, to be formed into fixed gases. Further, the gas generated from bituminous fuels is, due to condensible hydrocarbons, of a materially different composition before being cooled from what it will be after the cooling and cleaning process; there remaining vaporized in the hot gas the heavy hydrocarbons which condense at cooling.

The composition and heating-value of bituminous gas, when cold, does not in practice generally agree very well with any theoretical estimate based on the analysis of the fuel from which it has been produced. The percentage of hydrocarbons distilled over, and present in the gas, is a too important item that cannot be predicted, excepting, only approximately, by reference to some known analysis of a gas generated under conditions similar to those of the case in hand; and, further, the amount of hydrocarbons that becomes decomposed in the producer, into hydrogen and CO-gas, will depend very much on the temperature at which the gasification is made.

The reactions by which the hydro-carbons are broken up are principally the following:

$$CH_4 + H_2O = CO + 6H$$
$$CH_4 + CO_2 = CO + 4H$$
$$C_2H_4 + 2H_2O = 2CO + 8H$$
$$C_2H_4 + 2CO_2 = 4CO + 4H.$$

In the case of fuel-gas used hot from the generator, the heating-value, including the sensible heat of the gas, approximates often 90 per cent of the heating-value of the fuel, and the potential heating-value due to the hydrocarbons can in such a gas also be assumed to be 90 per cent of that of the hydrocarbons in the fuel. The heating-value of the hydrocarbons in bituminous coal

being generally from 17,000 to 20,000 B.T.U. per pound, we obtain, thus, in the hot gas, from 15,300 to 18,000 B.T.U. per pound of hydrocarbons in the coal used.

In the cooling and cleaning of the gas there disappears on an average from 50 to 70 per cent of the hydrocarbons, in the form of tar, and the loss in heating-value in the tar, with coals varying in volatile hydrocarbons from 20 to 40 per cent, will be from 10 to 24 per cent of the heating-value of the coal.

As a standard by which to compare the results actually obtained in practice, the ideal composition of the gas, estimated on a theoretical basis, becomes often of interest.

Assume the analysis of a sample of bituminous coal to be:

Fixed carbon	52 per cent
Volatile hydrocarbons	36 per cent
Moisture and ash	12 per cent

In a favorable case 25 per cent of the fixed carbon will be gasified into *C O*-gas by oxygen derived from steam, 65 per cent to *C O* by oxygen derived from the atmosphere and 10 per cent burned to CO_2-gas.

	Pounds of Gas.	Per Cent Weight.	Cubic Feet of Gas.	Per Cent, Volume.
Accordingly of 100 pounds of fuel:				
13 lbs. C gasified to C O with steam require 17.33 lbs. O, giving C O	30.33			
33.8 lbs. C gasified to C O with air require 45.06 lbs. O, giving C O	78.86			
TOTAL C O.....	109.19	30.1	1,480.6	26.6
5.2 lbs. C burned to CO_2 require 13.87 lbs. O, giving CO_2....	19.07	5.2	164.6	3.0
36.0 lbs. hydrocarbons distilled, giving C_2H_4, CH_4 and H.......	36.00	10.0	854.2	15.3
88.0 lbs. combustible.				
17.33 lbs. O dissociated from steam liberate H	2.17	0.6	412.3	7.4
58.93 lbs. O derived from the atmosphere carry N............	196.24	54.1	2,661.0	47.7
	362.67	100.0	5,572.7	100.0

The potential heating-value in the hot gas:

In 109.19 lbs. CO, at 4380 B.T.U. per pound......	478,252
In 36.00 lbs. C_2H_4, CH_4 and H, obtained from hydrocarbons, at 18,000 B.T.U. per pound ...	648,000
In 2.17 lbs. H at 62,100 B.T.U. per pound	134,757
	1,261,009

Heat energy in the gas from one pound of coal..12,610 B.T.U.
Heat energy in one pound of coal, hydrocarbons being figured at 20,000 B.T.U............ 14,800 B.T.U.

Efficiency of the conversion into gas, accordingly, 85.3 per cent.
Heating-value per cubic foot of gas, at 62° F.,....233 B.T.U.

If the average temperature at which the gases leave the producer be 1,000° F., and their specific heat 0.25, the sensible heat in 3.63 pounds of gas becomes 907 heat-units, which added to the potential heating-value of the gas brings the efficiency of the gasification into hot fuel-gas to 91 per cent.

In the cooling- and washing-process the gas would probably lose about 50 per cent of the hydrocarbons. The heat energy in the cold and cleaned gas becomes in that case:

In 109.19 lbs. CO at 4,380 B.T.U. per pound.......	478,252
In 18.00 lbs. C_2H_4, CH_4 and H obtained from hydrocarbons, at 18,000 B.T.U. per pound.........	324,000
In 2.17 lbs. H at 62,100 B.T.U. per pound.......	134,757
	937,009

The efficiency of conversion into clean gas, thus, 63 per cent. Heating-value per cubic foot of gas, at 62° F. = 168 B.T.U.

A theoretical analysis of bituminous gas, like the above, does not always agree very closely with practice, and it may be misleading if the true percentage of the fuel-value obtained from the hydrocarbons is not known from actual gas-analyses.

Hydrocarbon Loss in Tar.—The loss in heating-value in the tar can be determined through an analysis such as the following:

From an Indiana coal containing:

Fixed carbon 52 per cent

Volatile hydrocarbons 31 per cent

there was obtained a gas as per the volumetric analysis in the first two columns of the table below.

The resulting elementary constituents are as per the last five columns of the table.

Constituents.	Per Cent. Volume.	Molecular Weight.	Ratio of Weights.	Per Cent. Weight.	Elements.				
					N.	C.	O.	H.	CH_4.
N	51	28	1428	56.9	56.9				
CO	21.5	28	602	24.0	....	10.3	13.7	......	......
H	15	2	30	1.2	......	......	......	1.2	......
CO_2	9	44	396	15.8	......	..4.3	11.5	......	......
CH_4	3.5	16	56	2.1	......	......	......	......	2.1
			2512	100.0	56.9	14.6	25.2	1.2	2.1

The volatile hydrocarbons may, without great error, be assumed to be of the average composition of methane, CH_4.

Per 100 pound of gas:

Oxygen obtained from the atmosphere is $\frac{56.9}{3.33} = 17$ pounds.

Oxygen obtained from steam is $25.2 - 17 = 8.2$ pounds.

Steam used $= \frac{9}{8} \times 8.2 = 9.22$ pounds.

Hydrogen liberated from steam $= \frac{1}{8} \times 8.2 = 1.02$ pounds.

Hydrogen derived from the hydrocarbons $=$

$1.2 - 1.02 = 0.18$ pounds.

Hydrocarbons decomposed to form hydrogen and carbon $= 4 \times 0.18 = 0.72$ pounds.

Carbon liberated from the hydrocarbons $=$

$3 \times 0.18 = 0.54$ pounds.

Carbon derived from fixed carbon in the fuel $=$

$14.6 - 0.54 = 14.06$ pounds.

From 100 pounds of fuel, containing 52 pounds of fixed C, there is obtained $\frac{52}{0.1406} = 370$ pounds of gas.

The density of the gas is $\frac{25.12}{2} \times 0.0053 = 0.0666$ at 62° F. and 1 atmosphere.

Hence, 57 cubic feet of gas is obtained per pound of fuel.

In 100 pounds of gas there is accounted for 2.1 + 0.72 (= 2.82) pound $C\ H_4$, thus, per 100 pounds of fuel there is accounted for 2.82 × 3.70 = 10.43 pound $C\ H_4$, and there is lost 31 − 10.43 (= 20.47) pound of hydrocarbons in tar.

Hence the tar is 66 per cent of the hydrocarbon values.

Production of Water-Gas.—It is possible, in any gas-producer, to generate, intermittently, gas of a considerably higher heating-value than that which a continuous process will give. The gas production is manipulated as follows: The fuel-bed is blown up with pure air to a high temperature, while the lean gases generated during the blowing-up process are wasted to the atmosphere. If then, when the fire becomes of a sufficiently high temperature, steam, or, better, superheated steam, is blown through the fuel-bed it will be possible, for a short time, to gasify the steam and carbon without any air, resulting in a gas composed of practically only hydrogen and carbon monoxide. As soon as the fire becomes too cool for decomposing the steam the gasmaking process is discontinued, and the fire blown up again.

By proceeding in this manner, wasting the gases generated during the blowing-up process and collecting the gases formed by the decomposition of steam, the resulting product will be of a heating-value approaching that of the carbon monoxide or hydrogen, or approximately 324 B.T.U. per cubic foot. This gas is commonly referred to as water-gas. The sensible heat of the product generated during the blowing-up process, consisting mainly of carbon dioxide and nitrogen, does not actually need to be wasted, as it may be utilized in regenerators for preheating the air and super-heating the steam used for the process.

The composition of water-gas is, as may be seen from the reaction-formula, page 82, of the proportion of one pound of hydrogen in 14 pounds of carbon monoxide, or equal volumes of hydrogen and carbon monoxide.

When being used as illuminating gas, this gas is enriched by addition of hydrocarbons and illuminants, until a heating-value of about 550 to 600 B.T.U. is obtained. The gas is also used for industrial and power purposes. Illuminating gas obtained

in this manner is commonly referred to as carbureted water-gas or manufactured illuminating gas.

The original Siemens producer-gas, which is obtained without the use of steam in the producer, is occasionally designated by the name air-gas, and common producer-gas, being of a quality intermediate between air-gas and water-gas, is sometimes referred to as semi-water gas.

Gas-Producers.—Gas-producers are commonly classed with reference to the nature of the fuel with which they are intended to deal; as anthracite producers, bituminous producers, and lignite and peat producers.

The anthracite producers, which have to deal only with practically tarless and non-caking fuels, such as anthracite coal, coke, or charcoal, are generally arranged as suction producers. That is, the suction of the gas-engine piston, in connection with which the producer operates, is the only means whereby the draft through the generator is induced. This arrangement insures a very simple, compact, and easily handled installation.

The operation of the bituminous producers has, due to the nature of the fuel, more difficulties connected with it than the simple anthracite producer offers. These difficulties are mainly due to the caking of the fuel in the generator, which prevents the free flow of the gas-current through the fuel-bed, and also due to the formation of large quantities of tar in the process of cooling the gas.

The caking of the fuel is generally counteracted, as much as possible, by frequent poking operations, which have for object to keep the fuel-bed in a sufficiently porous state to allow the gases to pass, and escape freely.

To rid the gas of tar and dust there are employed a variety of static and mechanical cleaners, of which latter the centrifugal tar-extractor, illustrated in Fig. 174, page 451, has been found to be a simple and effective apparatus for cleaning any kind of bituminous gas and make it suitable for the gas-engine.

Anthracite Producers.—Minneapolis Suction Gas-Producer.—In Fig. 167 is shown a sectional view of a complete installation of an anthracite suction gas-producer of the type built by the

Minneapolis Steel and Machinery Co. The producer is fitted out with a pan-vaporizer, *W*, for generating the required quantity of steam. This vaporizer is built of steel as a ring-shaped pan, and it is placed on top of the furnace in such a manner as not to interfere with the poking of the fire through poke-openings, *P*,

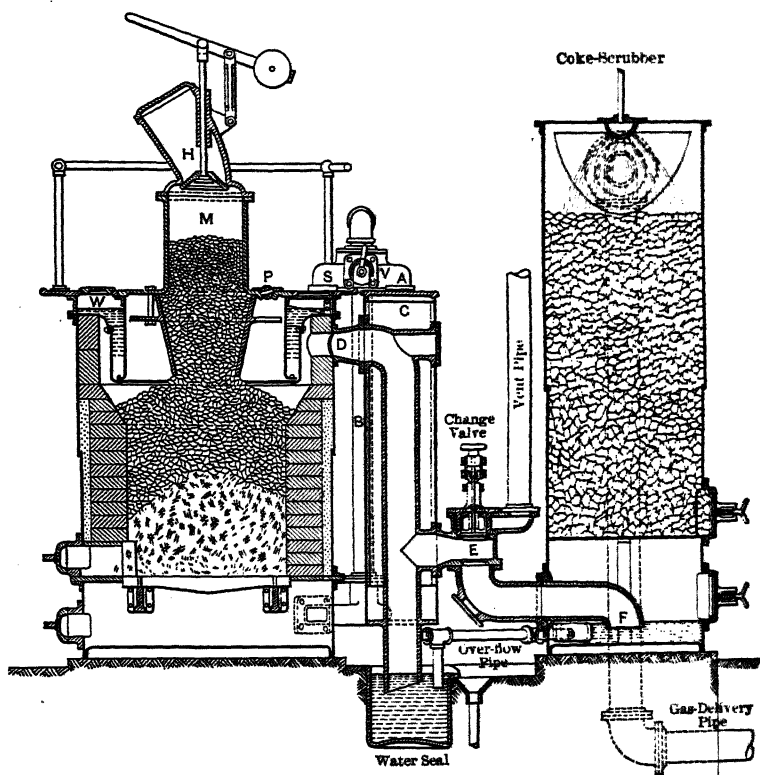



FIG. 167.—Suction Gas-Producer.

in line with the inside wall of the producer; it being important to be able effectively to remove, by poking, any cinders that tend to gather and build on to the walls. The gases escape around the outside of the waterleg of the pan to the gas-delivery pipe, *D*, which is carried down through an economizer, *C*, for pre-heating the air to be charged to the ash-pit. The sensible heat of the

escaping gases, is thus, well utilized, for generating vapor and for pre-heating the air.

A fuel-magazine, *M*, holding an ample supply of fuel, extends down through the centre of the water-pan, some distance, so as to bring the fuel-level to a proper height above the grate. The radiating heat from the top of the fuel-bed is taken up, partly by the fresh fuel-charge in the fuel-magazine, and partly by the inside surface of the water-pan. The heat-loss, due to radiation from the producer, becomes by this arrangement a minimum, and the producer top is kept cool.

A three-way valve, *V*, serves to regulate the proportion of air and steam furnished to the producer. The pre-heated air enters the valve at *A* and the steam at *S*, and by suitably adjusting the valve the proper proportion of air and steam will be delivered to the pipe, *B*, leading to the ash-pit. The cold air enters the economizer at *C*, and is carried down along one side of the gas discharge-pipe and up along its opposite side to the air admission, *A*, into the regulating valve.

E is a change-valve, which, when the fire is being blown up, is lowered so as to discharge the poor gases to the atmosphere, and when raised it discharges into the coke-scrubber, through the water-seal at *F*.

The coke-scrubber consists, as seen, simply of a steel-plate cylinder partially filled with coke, through which the gases pass slowly upward while water is delivered from a spray-nozzle above. Due to the large contact surface between the wet coke and the gases the latter will, in passing through the scrubber, become effectively cooled and cleaned from dust and tar.

From the coke-scrubber the gas is often carried directly to a small gas-tank, and from there to the engine. The object of the gas-tank is, partly, to separate and collect in it the water that may be carried along with the gas, but mainly to obviate in the producer, to some extent, the pulsations due to the periodical suction of the gas-engine piston.

Sometimes, there is placed in the gas-piping between the coke-scrubber and the engine a so-called dry scrubber, which apparatus is shown in the installation, Fig. 175. It consists of a steel-vessel

with a removable top, in which trays filled with shavings, or excelsior, are placed, through which the gas is filtered.

The trays of the purifier are, properly, made of wooden gratings, and the filtering material placed on them is often richly charged with iron turnings, which help to neutralize the acids formed by the oxidization of sulphurous fumes in the gases generated from fuels containing sulphur. The sulphur-compounds, H_2S and CS_2 in the gas are in themselves harmless to iron pipes and engine-parts, and act corroding first after oxidization to sulphuric acid. The conditions in the dry scrubber, the temperature, humidity, and presence of organic matter, are, however, very favorable for the oxidization of the sulphurous vapors, and it is in the apparatus itself the acids are formed for its own destruction. A heavy coating of iron oxide will help materially to preserve the metallic parts of the apparatus.

In order to effectively neutralize the acids, the trays in the dry scrubber are sometimes charged with a filtering material consisting of wood shavings, or excelsior, mixed heavily with iron oxide ore, so-called bog-iron ore. This material is generally used for the same purpose at gas-works and coke-oven plants.

Whenever a great percentage of sulphur is present in the gas any condensation of water-vapor in the engine should be carefully guarded against. The cooling, by jacket-water, of the exhaust valve-stem, piston or piston-rod and rod-packing should not be carried to such an extent as to cause the deposit of vapor on these parts, and it has, in some cases, been found expedient not to use cold water for cooling the piston, piston-rod, and rod-packing.

The practice of injecting water into the exhaust pipe, in order to cool the exhaust gas, is very injurious to the piping, whenever traces of sulphur are present in the gas, and should, as far as producer-gas installations are concerned, never be indulged in.

"Olds" Suction Gas-Producer.—A suction gas-producer of a somewhat different type from the one just described, one with an outside vaporizer, is illustrated in Fig. 168, together with necessary cooling and cleaning apparatus. The vaporizer, *W*, is here built on the principle of a water-tube boiler, and it is attached to, and independent of, the producer proper. It consists of a set

of water-tubes, which connect a lower water-chamber with an upper steam-chamber, and a surrounding flue through which the hot producer gases pass; being, in passing, deflected to the right and to the left, laterally to the axes of the tubes, by two baffle plates. The producer-gases are collected at the centre of the fuel-level in the producer by the pipe, *P*, which conducts them in to the vaporizer, and from the latter they are delivered, through the change-valve, *C*, to the cooling scrubber.

The apparatus illustrated, which is a 300-horse-power producer of the Olds design, contains several new features of interest. The air for the combustion is taken from the outside in to a pre-heating duct, *D*, surrounding the upper part of the furnace-shell, is heated by being carried once around the shell, and is delivered to the ash-pit through the pipe *A*. With the air is, of course, charged also the required quantity of steam, which is furnished by the vaporizer, through the pipe *S*.

The producer is equipped with a water-sealed, revolvable, top, upon which a water-sealed charging-hopper is mounted, a proper distance off from the centre, so that by revolving the top the fuel can be charged all around the outside of the fuel-bed.

The grate is revolvable and presents a conical surface against the fuel-body, with a central, adequate opening at the bottom, through which ashes and cinders are freely discharged.

The change-valve, *C*, is particularly of interest, as it is one of a water-sealed type. The casing of this valve consists, as seen, of a cylindrical vessel holding a small amount of water, through which the vent-pipe, *V*, passes up, a short distance above the water-level. The valve proper is mainly an inverted cup, which is dropped over the orifice of the vent-pipe, whereby it effectively seals the gas-passage against the access of any air from without. When this cup-valve is raised out of the water-seal, by means of the rack and pinion shown in the illustration, it carries with it, in a surrounding ring-cup, a quantity of water, into which the orifice of the pipe leading to the scrubber dips, and becomes, thus, properly sealed against access for air. In its raised position the valve leaves the vent-orifice free for discharge from the producer to the atmosphere.

The cleaning apparatus consists, as seen, of a combination wet and dry scrubber. The gas is delivered to the wet scrubber through the pipe *B*, and passing up through the water-sprayed coke-body it becomes cooled and cleansed of dust, after which it

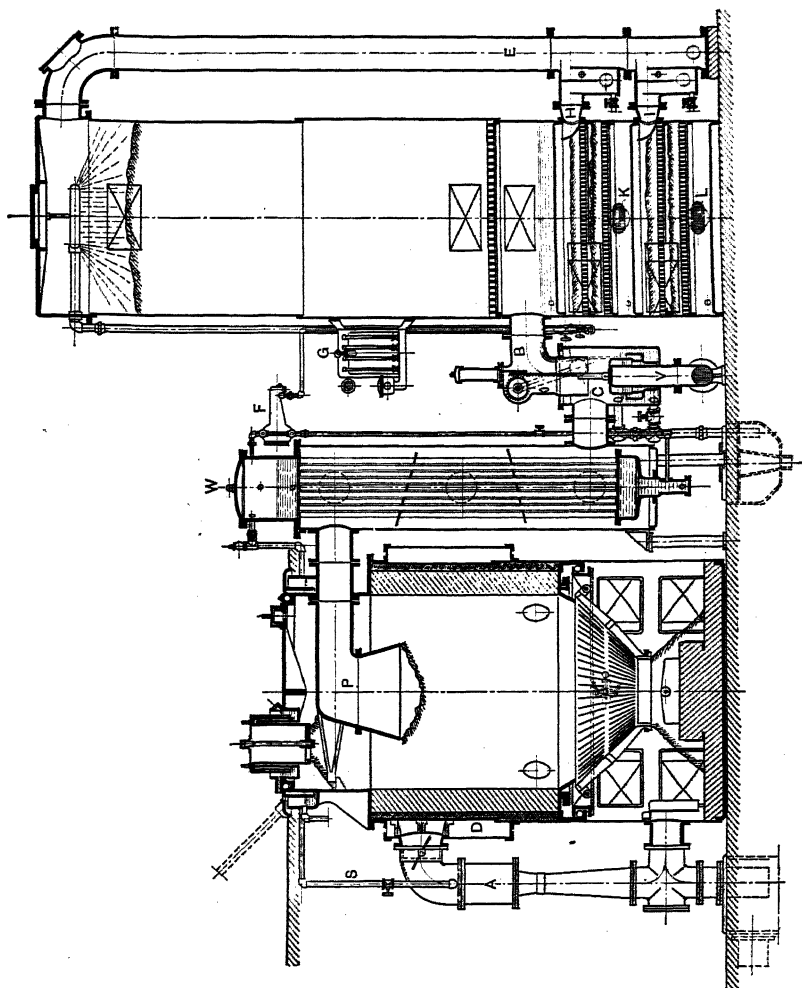



FIG. 168.—Suction Gas-Producer with Outside Vaporizer.

is passed, through the pipe *E*, in to a set of two dry scrubbers located in the base of the apparatus. The object of using two dry scrubbers is, simply, to double the filtering area through which

the total volume of gas must pass. The gas current divides itself, thus, between the upper and the lower filtering-chamber, passing in at *H* and *I*, and being delivered to the engine through the openings *K* and *L*.

G is a gauge-board with four water-gauges, each in order showing the gas-pressure, respectively, in the vaporizer, at the inlet side of the wet scrubber, at the outlet side of the wet scrubber, and at the outlet from the two dry scrubbers. By this means any obstruction in the gas-passages may readily be detected. A gauge-board of this kind, which makes it possible to ascertain quickly the pressure at different points of the gas-system, becomes, in connection with installations of some magnitude, a very necessary apparatus. A single water-gauge, piped up to the different points of the gas-system, with a small stop-cock in each branch may, however, answer the purpose.

F is a float-valve that controls, automatically, the water-level in the vaporizer.

Bituminous Producers.—The Water-Bottom Producer.—The water-bottom bituminous producer, Fig. 169, is a direct development from the original Siemens type. The grate for supporting the fuel is dispensed with, and the fuel-bed is made to rest on a column of ashes and refuse formed at the combustion; the whole fuel-column resting on the producer-bottom, which is made into a pan from which the refuse can very readily be removed, as required. The blast is carried to the centre of the column of ashes, and discharged by a distributing nozzle as evenly as possible over the full area of the bosh, some distance below the fuel-line. The producer is sealed by having the side-shell extended down into the ash-pan, which is luted with water.

This producer is used extensively for generating fuel-gas, and it may be said to be the standard type for fuel-gas. It is also, due to the fact that it involves a minimum amount of labor in its operation, used frequently for generating power-gas. It must then, however, be supplemented by effective cleaning apparatus for removing large quantities of tar.

More modern, bituminous producers are constructed and arranged with the idea of preventing the formation of any tar

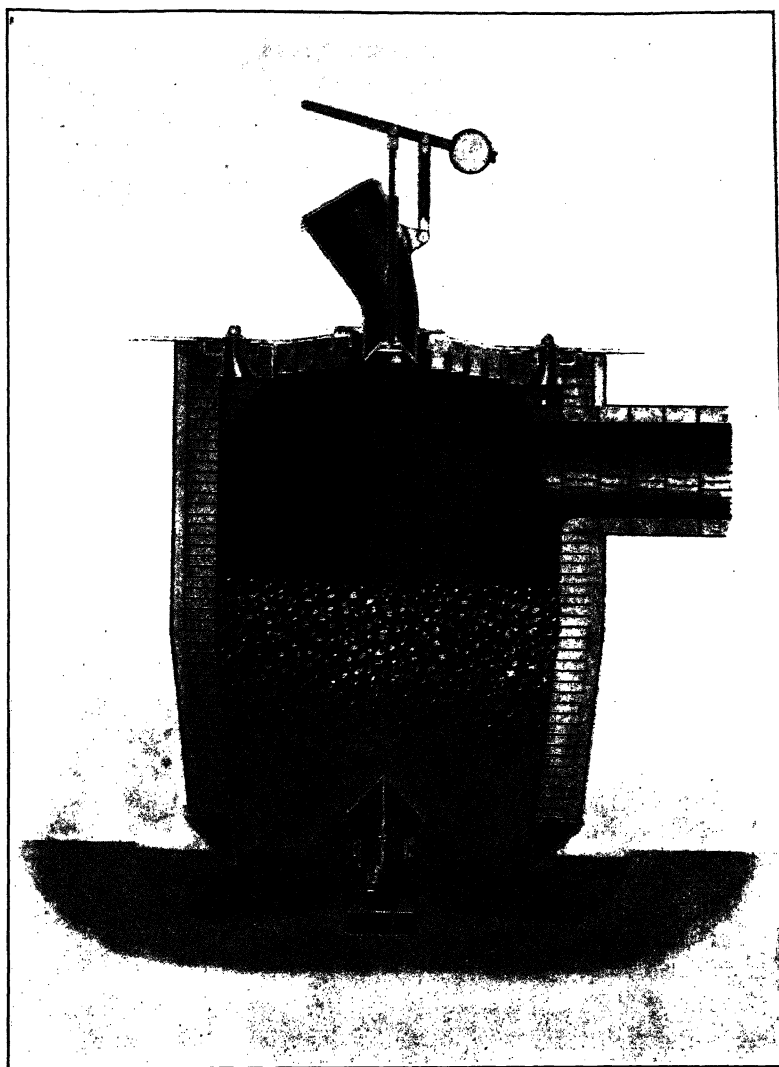



FIG. 169.—Water-Bottom Producer.

at the cooling of the gas. This is attempted by aiming to break up, in the furnace, into non-condensible gases, the hydrocarbons that form tar.

Tar is formed by the condensation of a variety of heavy hydrocarbons, many of which can readily be fixed into non-condensible gases by being heated to the required temperature. The new products which are formed at the decomposition of the hydrocarbons are hydrogen and carbon, or marsh-gas and carbon. Both these gases are stable and of high heating-value.

The Down-Draft Producer.—A producer built on this principle is the Loomis-Pettibone down-draft, double-furnace producer, a complete installation of which is shown in Fig. 170. In this producer the breaking up of the heavy hydrocarbons into incondensible gases is accomplished, in part at least, by carrying the gases which are distilled from the fresh fuel down through the incandescent coke in the lower part of the producer. The fuel will lose its volatile matter and become coked as it passes down into the furnace.

The producer proper consists, as will be seen, of two similar generators, 1 and 2, coupled to a common boiler, *F*, by means of the valves, *C* and *D*, so that they form one unit. A common exhauster, *E*, serves both generators for inducing the draft through their fuel-beds. Steam is introduced at the top of the generators and mingles with the air admitted through the top doors, *A* and *B*, which are open during the normal operation. The gases pass from the generators to the boiler for vaporizing the needed supply of steam, then, through the cooling and cleaning apparatus, to the gas-holder.

The fires may be regulated by passing, at regular intervals, a reversed current of steam through the fuel-beds for a fraction of a minute. This is effected by closing the openings, *A* and *B*, together with, for instance, the valve *C*, and admitting steam to the ash-pit of generator 1. The current, thus, passing up through the fuel-bed of this generator, then over to the top of generator 2 and down its fuel-bed, or *vice versa* if the valve *D* is closed and steam admitted to the ash-pit of generator 2.

The result of passing steam in this manner through tne fuel-

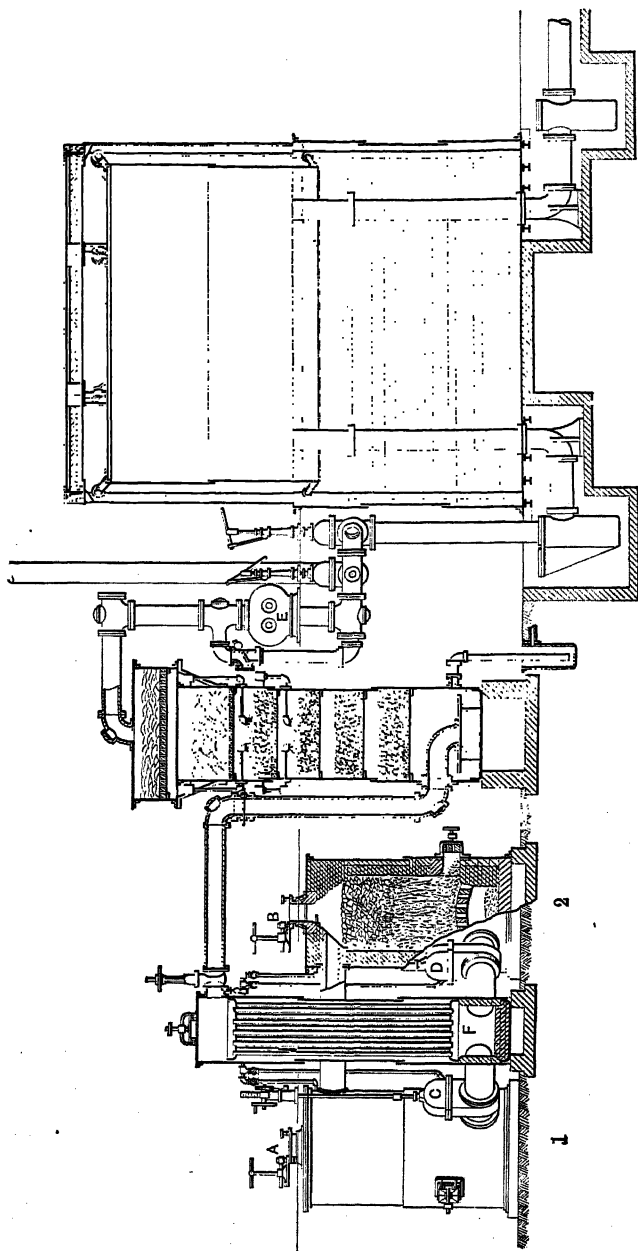



FIG. 170.—Loomis-Pettibone Gas-Generating Plant.

beds is the reduction of their temperatures, at the formation of water-gas. This gas being of a very high heating-value must be produced and mingled evenly with the gas normally generated, in order that any great fluctuation in the heating-value of the gas shall not be felt by the engine. To insure an even quality of gas, a gas-holder of some capacity is generally used in connection with this installation.

The gas-exhauster is connected to, and controlled by, the gas-holder, in such a manner that when the latter becomes filled with gas it shuts down the exhauster, thus stopping, for the time being, any production of gas.

The cleaning apparatus commonly used in connection with this type of producer, consists often only of a combination wet and dry scrubber, as shown in Fig. 170, supplemented by a saw-dust purifier which is not shown.

The wet scrubber consists of a cylindrical steel-plate vessel provided with several trays heaped with coke over which water flows from a spray nozzle above. The gas is admitted at the bottom, and ascends slowly through the coke-bodies and is due to contact with a large surface of wet coke cooled and freed from tar and dust.

Double Zone Producers.—The Deutz double-zone producer, Fig. 171, is a combination of a down-draft and an up-draft producer. It contains two zones of incandescent fuel. Air and steam are admitted into the upper part of the producer and the draft is downward. The hydrocarbons that are distilled off from the fresh fuel are thus drawn down through a bed of incandescent coke below, and fixed into a non-condensible gas, which is taken out through a delivery-pipe about midway of the height of the producer. At the same time there is another current from the ash-pit up through the lower zone of the producer. The main object with this latter current is to consume that part of the fuel that otherwise would pass out of the upper zone unconsumed. It helps also to maintain the upper zone in a high state of incandescence, thereby contributing to the thorough decomposition of most of the heavy hydrocarbons and to the reduction of the carbon dioxide gases formed in the upper part

of the producer. This producer appears to have many good points, as far as the production of a stable gas is concerned, but, in common with many of the bituminous producers, it requires excessive labor to keep the fires in proper condition, particularly

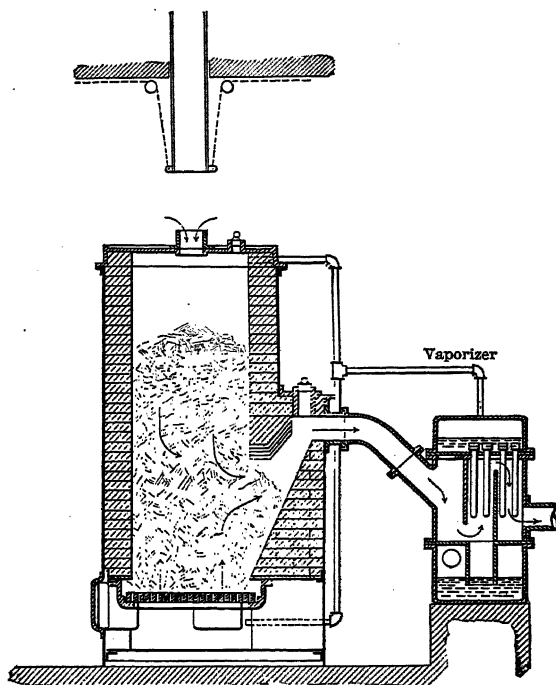



FIG. 171.—Deutz Double-Zone Producer.

if the fuel is not absolutely non-coking, and it is not very readily cleaned. In places in Europe it has, however, been very successfully used on lignites and briquetted peat.

Lignite and Peat.—The subject of lignite fuels for power purposes is at present of the very greatest interest, there being in the northwestern States, Dakota, Wyoming, and Montana, immense fields of this fuel available for use in the power gas-producer. The fuel is, however, of such a low grade that it cannot, under present conditions, be transported any distance with economy, and it becomes therefore mainly suitable only for home

consumption. It will, on this account, be necessary to locate the gas-producer plant conveniently to the fuel-supply.

In the following table are given the composition and heating-value of samples of lignite from different localities, and, for the purpose of comparison, the analyses of two other fuels, wood and peat, that are occasionally used in the producer.

The heating-values of wood and peat are those of the perfectly dried fuels, the moisture in the air-dried samples having been allowed for.

The heating-values of the American lignites are those given by the analyses of air-dried samples, and, the second figure, those corresponding to perfectly dried samples.

When drying in the air, the lignites lose from 10 to 30 per cent of their original water contents.

	Wood. Average.	Peat. Average.	Lignite.	
			Alameda Co., California.	Stark Co., N. Dakota.
Moisture in air-dried fuel			18.5	32.6
Volatile matter			35.3	29.2
Fixed carbon			30.7	26.8
Ash			15.5	11.4
Ash		6.0	15.5	11.4
Sulphur			3.0	3.5
Hydrogen	6.2	6.0	5.9	6.2
Carbon	50.0	56.0	47.4	39.5
Nitrogen	1.0	1.0		
Oxygen	41.3	31.0	27.5	38.9
Calorific value as per analyses			8105	6970
Calorific value, moisture allowed for				10800

An endurance test of a Deutz two-zone, down-and-up draft producer on lignite fuel, lasting 320 hours, showed a fuel consumption per brake horse-power of 1.46 pounds.

An analysis of the fuel gave:.

A calorific value of 8,500 B.T.U.

Fixed carbon	35.0 per cent.
Volatile matter	45.0 " "
Ash	4.8 " "
Water	15.2 " "

The resulting gas showed no appreciable amount of tar.

Another test of this producer showed at heavy load (100 B.H.P.), a fuel consumption of 1.1 pound.

The analysis of the fuel gave:

Fixed carbon	42.0 per cent.
Volatile matter	31.2 " "
Ash	4.4 " "
Water	22.45 " "

The Jahn Producer.—The Jahn producer, Figs. 172 and 173, which in Germany is working very successfully on any grade of cheap fuel, consists of four up-draft furnaces or retorts connected by means of valves, in such a manner that the current of gas can be shifted from one furnace of the set in to any other one of the same set. The furnaces are charged at regular intervals, starting with number 1, next number 2, and so on, in succession. The gases from the latest charged furnace, number 4, which to a great extent consist of hydrocarbons and moisture distilled off from the fresh fuel, are drawn through the fuel-beds of the second and third units. By this means, it is quite apparent, the gases, when they leave the system, must be of a highly stable quality, they having been passed through the incandescent coke-beds of the second and third units. Furnace number 1 of the set is, in the mean time, switched out of the system for being cleaned, re-charged and put in readiness to take the place of unit number 4, in due time.

The general construction of the producer may be studied by reference to Figs. 172 and 173. The upper half of Fig. 172 represents a plan view looking from the top of two furnaces, and the lower half is sections taken through the furnaces, at *C C* and *D D*. The left-hand half of Fig. 173 represents a vertical section

taken on the line A A, and the right-hand half is a section taken, on the bias, through the line B B.

It will be seen that each individual furnace communicates, through ports E and F, with a central gas-flue, G, respectively, above the main part of the furnace and at the bottom of the furnace. These passages can be opened and closed according to requirements by valves such as I and J, which are all operated from the charging floor. The main shafts of the furnaces terminate, above the charging floor, in sealed hoppers, H, through which the fuel is charged, and through which also the fixed producer-gases are conducted to the gas-main.

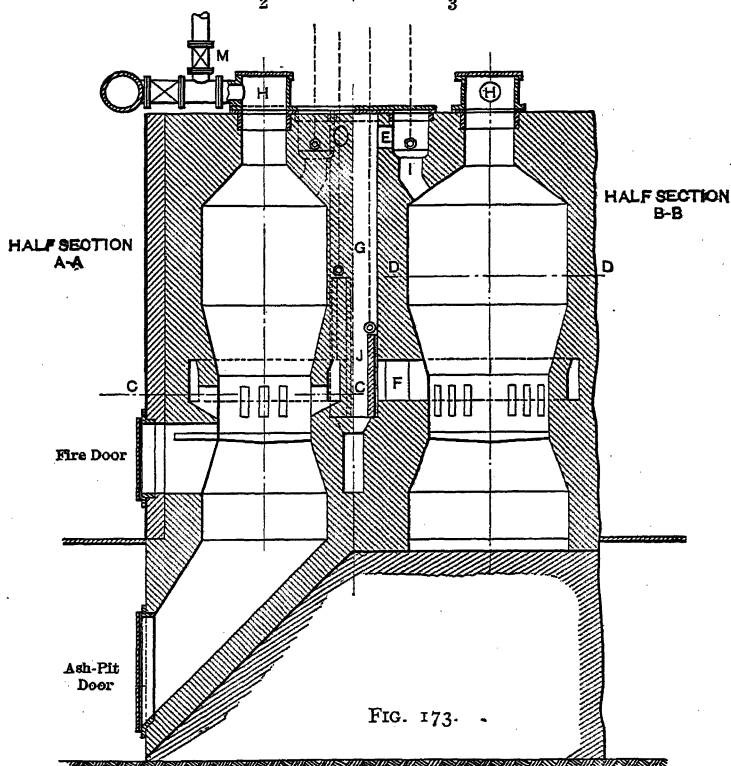
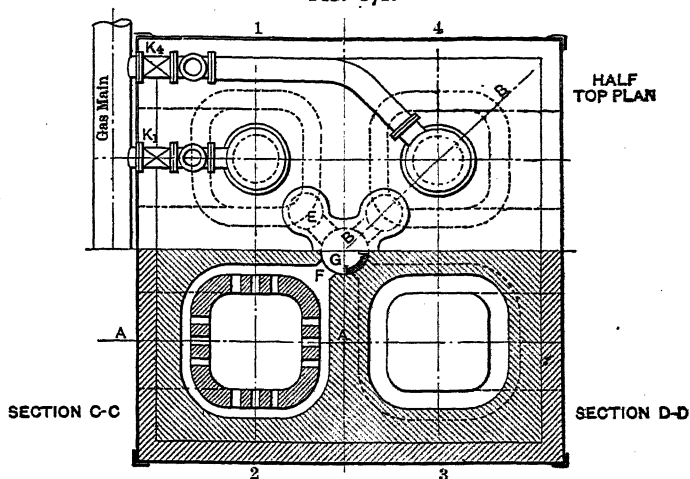
As each one of the hoppers, H, will, in turn, be opened at times when its furnace is being cleaned and re-charged, there must, of course, be provided a gas-valve in the pipes communicating between the hoppers and the gas-main. These valves, for furnaces 1 and 4, are shown at K_1 and K_4. It will be necessary, at times when the furnaces are first started up, to get rid of some gas too poor for being used, and for that purpose there is also provided for each furnace a vent-valve, which is shown at M.

The gas which is fixed and ready for the engines is, by means of an exhauster, drawn from the furnaces through suitable cooling and cleaning apparatus, and delivered to a gas-holder.

Assuming that the furnaces have been put in normal operation, and that furnace number 4 has been switched out of the system for cleaning and re-charging. The valves I and J are then, as shown in the drawing, closed, as is also the valve K_4 in the gas-pipe leading from hopper of furnace 4. This hopper, as well as the fire- and ash-doors of furnace 4, can then be opened to facilitate the cleaning of the furnace, because the furnace is isolated from the rest of the system. When ready to be switched in to the system, the hopper and fire-door of furnace 4 are closed and the valve I opened; the valve J and the gas-valve, K_4, remaining closed. The draft current will, thus, be: From the ash-pit through the fuel-bed of furnace 4, then through the port E in to the central gas-flue, G. From there it will pass through the passage F in to the incandescent coke-beds of furnaces 2 and 3 to be drawn from their hoppers to the general gas-main;

FIG. 172.

FIG. 173.

Jahn Producer.

furnace 1 being supposed to have been in the mean time, switched out of the system to be cleaned.

The furnaces are built of capacities to hold a charge of three to four tons, and may be arranged as many as required in one group; each four furnaces being one set. The brickwork must be laid in a substantial manner, and the whole group of furnaces is encased in a steel casing.

A successful power-gas plant, containing several of the Jahn producers, has been in use, since 1902, at the Van der Heydt coal mines in Germany, the fuel being refuse containing only 20 per cent coal. The producer seems to possess, in a promising manner, all the requirements for a continuous production of a fixed gas from low-grade fuels, excepting one—convenience for poking the fires to keep the fuel-beds open, and it has been learned that coal that has the tendency to coke is unfit for use in this producer.

Gas-Washers.—A centrifugal gas-washer of efficient type is shown in Fig. 174. It consists of a horizontal drum fitted with radial vanes, which is journaled in a gas-tight steel casing also carrying vanes that clear those on the drum by a small margin; one-half to three-quarters of an inch. The drum is revolved at a suitable speed; and the gas, after having been cooled in some kind of cooling apparatus, is admitted to the washer through the pipe, *P*, and in passing between the stationary and revolving vanes of the apparatus is thoroughly mixed with a spray of water admitted through the pipe *W*. The tar-products and water are beaten up to an emulsion which passes off along the shell of the casing to the bottom of the washer; collecting in the lower settling-chamber, and passing out at *D* to the drain-tank, *T*.

The current through the apparatus is impelled by the vanes *I* at the outlet end of the revolving drum, and the gas leaves the discharge-chamber, *E*, at a slight pressure. If the pressure in the discharge-chamber should, due to a decreased gas consumption, increase a small amount above that contemplated, then the excess of gas will become by-passed through the water-seal at *S* and be returned through the ports *R* to the inlet side of the washer. There is provided a water-seal also at the bottom of the

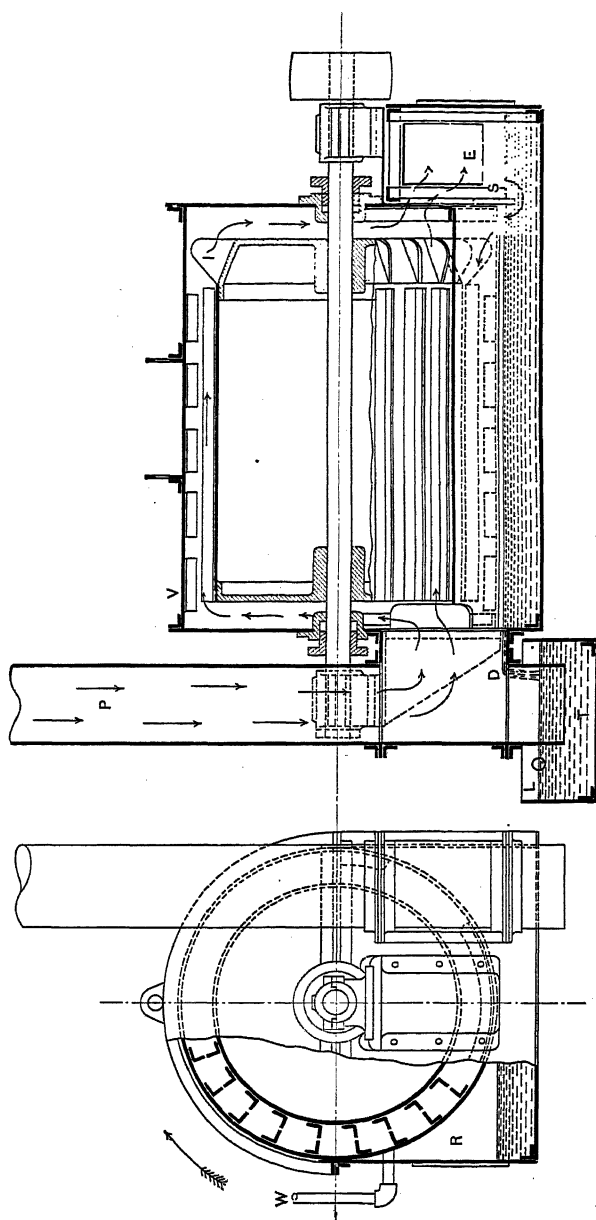



FIG. 174.—Centrifugal Gas-Washer.

inlet gas-pipe to afford a relief for the pressure in case an explosion should occur in the apparatus.

The light tar-products will be drained off from the drain-tank at *L*, whereas heavy products will be collected and drained from the bottom of the tank, or from the bottom of the settling-chamber of the washer. An effective apparatus of this type will readily remove the tar in the gas to such an extent that the impurities remaining shall not average more than 0.015 to 0.02 grains per cubic foot; the gas thus being amply clean for any purpose for which it may be used.

A washer handling the gas from a 600 horse-power producer would normally require 15 to 20 horse-power, but, as some overload capacity will be required in starting, a 25 horse-power engine or motor would suitably be provided. The water-consumption will not exceed two gallons per horse-power per hour.

Capacity of Producers.—The rate at which coals may properly be gasified in the gas-producer is at present a more or less unsettled question, due to the variety in the character of the coals generally used. It has frequently been held by German authorities, that in an anthracite producer seven to nine square inches of sectional area of the fuel-column would be a proper allowance per horse-power, but the practice with American anthracite coal has shown that ten to sixteen square inches give far better results on a somewhat steady load. When the load-conditions are such that the engine may run very light for a considerable length of time (say, less than one-quarter of full load for about twenty minutes), and then the full load be suddenly thrown on, and if such changes are liable to occur repeatedly, then it is advisable to install a relatively small producer. Since the large producer is unnecessarily large at light loads, it will, if the vapor-supply is not carefully controlled, become cooled down during the light load to such an extent that when the heavy load comes on it may not be in a condition to respond with the required supply of gas. The smaller producer, on the other hand, will not, under reasonable short intervals of heavy load, have time to become heated enough to give trouble on account of a clinkering fuel, and will maintain better a normal condition during light-load

periods. When the load is steady a large producer is, however, to be preferred.

When a producer is used for generating gas for power purposes its capacity is generally expressed in horse-power, although it would be more correct to express it by the number of pounds of coal it can gasify per hour. If the fuel-consumption be counted 1¼ pound of coal per horse-power, then 10 to 16 square inches sectional area per horse-power would correspond to a gasification of 18 and 11 pounds of fuel per square foot fuel-area per hour.

In a bituminous producer good gas can be generated from suitable coal at the rate of 20 pounds of fuel, or more, per square foot, but with less suitable fuels the gasification should not be driven at a higher rate than 10 pounds per square foot; the best rate for gasification being very much dependent on the quality of the fuel.

The reaction in the producer whereby good gas is produced depends on the amount of hot fuel-surface exposed to the action of the gases passing through the fuel-bed, its temperature, and the speed with which the gases pass. If the current is forced through at a high rate of speed the temperature of the fuel-bed must be high in order to reduce the $C\,O_2$-gas properly, and the hot zone becomes high, wherefore the gases pass off hot. If, in such a case, the fuel is of a good quality, containing a low percentage of ash and refuse of a high fusing-point, then the production of good gas will proceed with no other inconvenience than that there will be incurred some loss due to the excessive amount of sensible heat carried off by the gases. On the other hand, should the fuel be of a low grade, containing a great amount of highly fusible refuse, then great inconvenience will be experienced on account of the formation of clinkers that impede the even flow of the gases over the whole area of the fuel-bed. The current will seek channels forming through the fuel-bed and gas of a poor quality will escape. A continuous poking of the fire to remove clinkers must in this case be resorted to, in order to keep up the gas-production.

With low-grade fuels, the operation of the producer, at the rate of 10 pounds of fuel per square foot fuel-area and at a tem-

perature of the fuel-bed of 1,800 to 2,000° F., may be continued without any great inconvenience from the fusing of the ash, whereas at an increase of 25 per cent of the output the temperature of the furnace may rise to 2,200 degrees and cause a great expenditure of labor to keep the operation going.

With regard to low-grade fuels containing a great amount of highly fusible ash, the nearest to a correct statement as to the rate at which it should be gasified may, perhaps, be to say, that the proper rate is only that which involves the least amount of labor for producing a gas of good quality.

Size of the Gas-Outlet Pipe.—The following may serve as a guide for determining the size of the gas-outlet pipe of a power-gas producer. There is generated, in the anthracite suction producer, about 5½ pounds of gas per pound of coal, or, in volume, on an average, 80 cubic feet at 62° F. Assuming the gas to leave the producer at 900 degrees, its volume at that temperature will be 208 cubic feet per pound of coal, or, per pound of coal gasified per hour, there will be discharged 0.06 cubic feet per second.

Limiting the mean speed of the gases through the discharge-opening to 30 feet per second, the required area of the opening becomes 0.28 square inches per pound of coal gasified per hour. There is generally figured, as a safe allowance, 1¼ pound of anthracite fuel per brake horse-power per hour, and, hence, the required discharge opening per horse-power would be 0.35 square inches.

For the type of anthracite producer shown in Figs. 167 and 168 this area of discharge pipe is ample.

The gas from a bituminous producer is often of a higher temperature than 900 degrees, it being less thoroughly cooled by the fresh fuel-charge and vaporizer, but, on the other hand, the volume of the gas generated per pound of fuel is somewhat less than that of anthracite gas. An area of the discharge opening of 0.28 square inch per pound of fuel gasified per hour, or 0.35 square inch per horse-power, will therefore be ample even for the bituminous power gas-producer. When cooled, the volume of the gas is less than one-half the volume at 900 degrees, wherefore

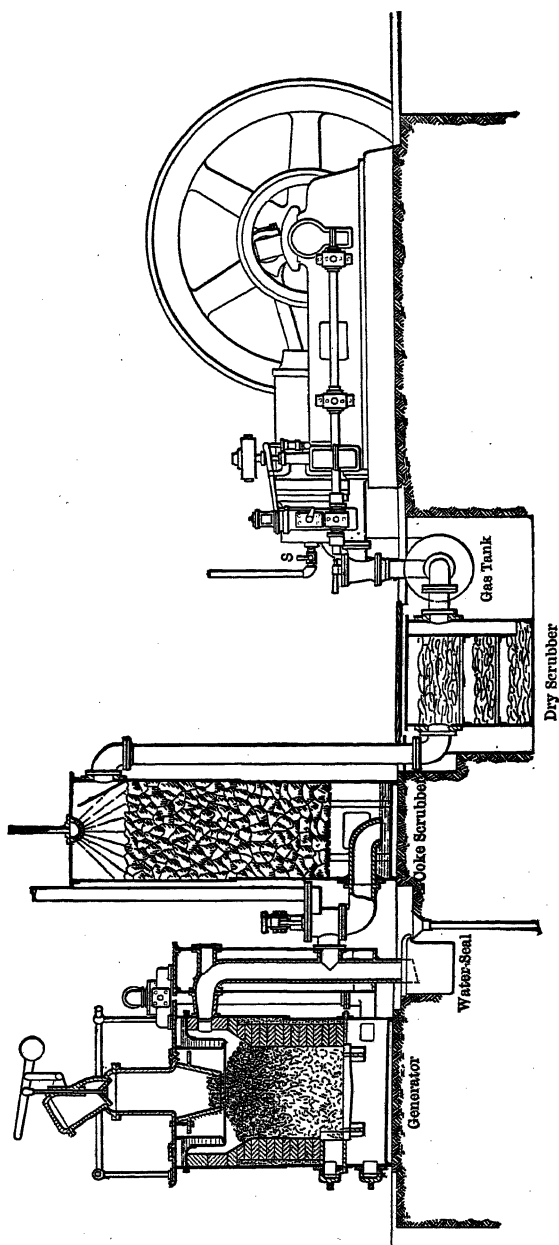



FIG. 175.—Suction-Gas Power Installation.

the pipe carrying the cool gas can, consistently, be smaller than the hot gas-pipe.

For a fuel-gas producer the discharge pipe would be larger than the proportions given, which apply for power-gas producers. Figuring the maximum capacity of a fuel-gas producer 18 pounds of fuel per square foot area, and the discharge-opening 0.5 square inch per pound of fuel gasified, the discharge-pipe would be nine square inches in area for every square foot area of the producer, or at a ratio of 1 to 16. The diameter of the discharge-pipe, therefore, one-quarter of the diameter of the producer.

Producer and Gas-Engine Installation.—Fig. 175 illustrates a complete installation of a suction gas-producer and engine. In the way of cleaning apparatus the installation contains the usual wet scrubber (coke-scrubber), and a dry scrubber which, in order to economize on space, is placed below the floor-line. The engine is started by means of compressed air which is delivered from an overhead air-tank to the starting-valve, *S*. An air-pressure of 120 to 140 pounds is commonly used. The size of the air-compressor, usually driven by a small gasoline- or gas-engine, and the size of the air-tank, should be suited to each other so that the time for storing up the pressure does not become excessive.

For all single engines of Table XXV a 3 x 4 single-acting, two-cylinder compressor running 200 to 250 turns will be adequate, and the same compressor will do for twin or tandem engines up to a size of 17 x 28. For any twin or tandem engine, referred to in the table, above that size a 3¾ x 5 compressor will be better.

The complete installation must also include a pump for circulating the cooling-water through the jacket of the engine. The required size of such a pump may be determined according to the method given at page 297.

APPENDIX

Gas-Engine Tests.—In the following are recorded, in detail, a few gas-engine tests of acknowledged high efficiencies, comprising various kinds of fuels. The main data relating to these tests have been included in Table XXXI, and below will be found, in carefully tabulated form, the detail data of interest as given by the experimenter, but transcribed to the English system of weights and measures.

TEST OF A 500-HORSE-POWER BORSIG-OECHELHAEUSER ENGINE on coke-oven gas, at Silesia, Germany. Test by Prof. E. Meyer of Charlottenburg, Oct. 26, 1903.

The main dimensions of the engine, which is of the two-cycle type shown in Figs. 134*a* and 134*b*, are:

WORKING CYLINDER.		
Diameter	26.6	inches
Length of stroke	37.5	inches
BLOWING TUB.		
Diameter	65.0	inches
Length of stroke	37.5	inches
Diameter of piston-rod	6.0	inches
DOUBLE-ACTING AIR PUMP.		
Diameter of piston	45.0	inches
Length of stroke	19.6	inches
Diameter of piston-rod { front	3.54	inches
Diameter of piston-rod { rear	2.75	inches
SINGLE-ACTING GAS PUMP.		
Diameter of piston	23.2	inches
Length of stroke	19.6	inches

The composition of the gas used is shown by the following table:

Analyses of Coke-Oven Gas.

TEST NUMBER	I.	II.	III.
Time	10 a.m.	4 p.m.	10 a.m.
PER CENT HYDROGEN, H.	42.00	48.08	43.80
Carbon monoxide, CO	11.84	10.60	10.20
Heavy Hydrocarbons	2.63	1.80	2.10
Methane, CH_4	19.73	18.43	20.30
Nitrogen, N	18.69	15.89	17.90
Carbon dioxide, CO_2	4.91	4.90	5.30
Oxygen, O	0.20	0.30	0.40

TABLE XXXII.

Data from Test of October 26, 1903, of a 500 H. P. Borsig-Oechelhaeuser Engine, Working on Coke-Oven Gas.

Number of test	VIII.	IX.	X.	VI.	VII.
Duration of test—minutes	20	15	40	15	20
Speed of engine—revolutions per minute	103.0	107.0	106.1	108.2	107.4
Working cylinder — M.E.P.—lbs per sq. in.	75.1	73.8	69.3	62.3	62.0
Working cylinder — Total indicated work = I.H.P.t	810	827	769	705	697
Blowing cylinder — Total indicated work, or brake horse-power of the engine = B.H.P.	616.2	627	575	488	474
Air pump — M.E.P., in front of piston—lbs.	5.09	5.38	5.12	5.56	6.09
Air pump — M.E.P., back of piston—lbs.	3.36	3.58	3.41	3.73	3.94
Air pump — Indicated H.P. Consumed = W_a	68.3	75.2	71.1	79.0	84.5
Gas pump — M.E.P.—lbs.	3.53	3.50	3.58	3.73	3.84
Gas pump — Indicated H.P. consumed = W_g	7.7	7.8	7.9	8.5	8.7
Total indicated H.P. consumed by charging pumps = $W_a + W_g$	76.0	83.0	79.1	87.5	93.2
Net indicated work of engine = I.H.P.n = I.H.P.t − W_a − W_g	734	744	690	617	603
$\frac{\text{Total pump work}}{\text{Net ind. work}} \times 100 = \frac{W_a + W_g}{\text{I.H.P.n}} \times 100$—%	10.3	11.1	11.4	14.2	15.5
Total efficiency between working cylinder and blower = $\frac{\text{B.H.P.}}{\text{I.H.P.t}} \times 100$—%	76.1	75.8	74.8	69.2	68.0
Mechanical efficiency of engine = $\frac{\text{B.H.P.}}{\text{I.H.P.n}} \times 100$—%	83.9	84.2	83.3	79.1	78.6
Work consumed in friction in the engine = I.H.P.n − B.H.P.	117.8	117.	115.	129.	129.
Gas of 32°F. and 760 m/m — consumed per hour—*c. f.*	13505	13951	13198	12100	11800
Gas of 32°F. and 760 m/m — lower heating-value B.T.U. per *c. f.*	400.1	394.5	383.3	394.5	397.9
Heat consumption — Total per hour—B.T.U.	5,505,000	5,504,000	5,060,000	4,774,000	4,694,000
Heat consumption — per total I.H.P.—B.T.U.	6679	6639	6599	6760	6720
Heat consumption — per net I.H.P.—B.T.U.	7364	7404	7324	7726	7766
Heat consumption — per brake H.P.—B.T.U.	8772	8772	8772	9778	9899

This engine test is notable, not only on account of the high efficiency it records, but also on account of the unusually low average heating-value per cubic foot of actual charge.

Figuring the capacity per unit volume of the cylinder at par with that of a non-scavenging engine (the suction displacement-volume thus equal to the volume of the actual charge) we find:
The suction displacement-volume per revolution to be 24.1 cubic feet;
The suction displacement-volume, per minute, of test X, 2557 cubic feet;
The total indicated power of test X is 769 I.H.P.;
Hence, the suction displacement per ind. horse-power is D = 3.32 cubic feet.
The thermal efficiency of test X is $Efy = 0.385$ with respect to the indicated output.

If the above quantities are inserted in formula 52 we obtain:

$$\frac{H}{\frac{V_a}{V_o}\left(xa + 1\right)} = \frac{42.42}{Efy\ D} = \frac{42.42}{0.385 \times 3.22} = 33.2.$$

Thus, referring to test X, the heating-value per cubic foot of actual charge is 33.2 B.T.U. only; and with respect to the other tests of the series it will be found also to approximate this figure.

The assumptions made in Chapter VI, with respect to the normal charge were the following:
The excess air is 15 per cent of that theoretically required, the pressure of the charge at completed suction-stroke is 13.2 pounds absolute, and its temperature is 120° F. Under these conditions the normal charge would be of a heating-value of 61 B.T.U. per cubic foot. Hence, the preceding test would indicate that for obtaining the best efficiency the required excess air-charge should be much greater than the assumed 15 per cent allowed in the normal charge. It is true that many efficiency tests on illuminating- or coke-oven gas have given excellent results with an excess dilution of 30 to 50 per cent above that theoretically required, but, then again, many tests show poor results when a too highly diluted charge has been used, and it is exceptional that a very

high efficiency is obtained with a charge diluted, as in the above test, to approximately twice the volume of that which in the previous has been designated the normal charge.

As an explanation of the high efficiency obtained in the above case, in spite of the highly diluted charge used, the circumstance that the hot neutrals are not fully expelled from the cylinder would probably be quoted; and that they are separated from the fresh charge, which therefore, actually, contains essentially less air than appears from the above computation. In that case the neutrals remaining in the cylinder serve in the main only to reduce the effective cylinder-volume.

For the sake of comparison with the above test, it will be of interest to make some deductions from a test of another two-cycle engine of high economy.

The test of a Koerting engine at Hanover, recorded in Table XXXI, gave on producer-gas an efficiency $Efy = 0.34$. The cylinder dimensions are 21.6×37.7 inches, and the speed of the engine was 101 revolutions per minute.

Assuming the piston-rod to be of a diameter of 5 inches, we obtain:

The suction displacement per stroke 7.55 cubic feet, and the suction displacement per minute 1525 cubic feet.

The work generated was 481 I.H.P., and, hence,

the suction displacement per indicated horse-power per minute 3.17 cubic feet.

The heating-value of the actual charge becomes, therefore, per cubic feet:

$$\frac{H}{\left(xa+1\right)_o} = \frac{42.42}{0.34 \times 3.17} = 40 \text{ B.T.U.}$$

The low heating-value of the gas used being 129 B.T.U. per cubic foot, the heating-value per cubic foot of normal charge would probably be in the neighborhood of 45 B.T.U.

In this case we have, therefore, a two-cycle engine which gives a high efficiency, using a charge much less diluted than in the case of the previous test.

In the case of the test of the Premier engine at Winnington, also recorded in Table XXXI, a thermal efficiency referred to the indicated output of 0.337 was obtained. This engine, of four-cycle scavenging type, has a suction displacement of 10.61 cubic feet per revolution, which is equivalent to a suction displacement, at the test, of 2.78 cubic feet per indicated horse-power per minute.

The heating-value per cubic foot of charge was, accordingly, at the test 46 B.T.U.; whereas the heating-value of the so-called normal charge (the gas being of 144 B.T.U.) would probably approximate 48 B.T.U.

Again, the test of the double-acting Nuernberg blast-furnace gas-engine at Rombach, recorded in Table XXXI, gave a thermal efficiency $Efy = 0.339$, and the suction displacement per indicated horse-power per minute was $D = 3.1$ cubic feet. The heating-value per cubic foot of charge, accordingly, 40 B.T.U., whereas the heating-value of the so-called normal charge (the gas being of 88 B.T.U.) would probably be somewhat less than 40 B.T.U.

From the above deductions it will appear, thus, that it is not generally necessary for obtaining a good economy that the charge should contain a very great excess of air, though it is possible that an apparently highly diluted charge may give a very ample efficiency, if the neutrals in the cylinder can be kept separated from the actual charge.

TEST OF A 300 HORSE-POWER HIGH-SPEED DIESEL ENGINE at the Augsburg Works of the Diesel Co. The test made by Chr. Eberly, of Munich, and recorded at page 180 of the Zeitschrift des Vereins Deutscher Ingenieure, February 1, 1908.

The fuel used was Galician crude oil of the following composition:

Carbon	86.41 per cent
Hydrogen	12.66 per cent
Sulphur	0.85 per cent
Oxygen and nitrogen	0.08 per cent
	100.00

Lower heating value, 18,130 B.T.U. per pound.

TABLE

Data from Test of a 300-

Number of test	1		2	
Duration of test—hours	1.16		1.18	
Speed of engine—revolutions per minute	256.8		306.6	
Indicated power of working cylinder less work consumed by air-pump—I.H.P.	239		292	
Brake horse-power of engine—B.H.P.	197		233	
Mech. efficiency = $\frac{\text{Brake H.P.}}{\text{Ind. H.P. of work. cyl.} - \text{Air-pump work.}}$	82.5		80.0	
Fuel consumption per hour: per ind. H.P. neglecting air-pump work—lbs	0.322		0.316	
Fuel consumption per hour: per brake H.P.—lbs	0.420		0.426	
Fuel consumption per hour: per brake H.P. referred to fuel of 17600 B.T.U. per pound, lower value—lbs	0.431		0.437	
Cooling water used per brake H. P. per hour—lbs	69.5		58.5	
Temperature of cooling water: at inlet—°F	55		55	
Temperature of cooling water: at outlet—°F	93		100	
Temperature of exhaust gases—°F	624		696	
Analyses of exhaust gases: contents CO_2-gas—%	6.8		8.3	
Analyses of exhaust gases: contents oxygen—%	8.4		....	
Heating-value of the fuel B.T.U. per lbs.	18126		....	
Heating-value consumed: per ind. H.P.—B.T.U.	5842		5738	
Heating-value consumed: per brake H.P.—B.T.U.	7617		7718	
HEAT DISTRIBUTION PER ONE POUND OF THE FUEL.	B.T.U.	%	B.T.U.	%
Converted into indicated work	7893	43.5	8037	44.3
Converted into brake horse-power	6057	33.4	5967	32.9
Friction, and work consumed by the air-, oil-, and water pumps	1836	10.0	2070	11.4
Dissipated into the cooling water	6210	34.3	6066	33.5
Carried off by the exhaust gases	4374	24.1	4140	22.9
Discrepancy	−351	−1.9	−117	−0.7
Total = lower heating value of the fuel	18126	....	18126	....

XXXIII.

Horse-Power Diesel Engine.

3	5	7	8	9	10	11	12
0.83	0.85	0.53	0.42	0.49	0.46	0.53	0.60
402.4	497.4	301.1	247.4	400.1	488.1	400.5	508.1
386	402	181	146	221	246	393	....
294	335	126	106	156	165	322	390
76.2	83.3	69.6	72.6	70.6	67.2	81.8	....
0.306	0.337	0.284	0.292	0.302	0.334	0.328	....
0.435	0.454	0.453	0.448	0.483	0.560	0.439	0.460
0.447	0.466	0.464	0.460	0.496	0.574	0.451	0.48[illegible]
58.3	57.4	....	....	....	....	....	....
55	55	....	....	....	....	....	....
99	104	....	....	....	....	....	....
734	826	432	401	496	617	762	....
8.4	9.4	4.5	4.4	4.6	4.8	9.6	....
8.0	7.0	13.8	14.0	13.2	12.6	8.4	....
....	....	....	....	....	....	....	....
5553	6121	5142	5292	5477	6064	5963	....
7903	8225	8205	8124	8772	10152	7963	8531

B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%	B.T.U.	%
8298	45.8	7524	41.5	8962	49.4	8712	48.0	8415	46.4	7605	41.9	7731	42.6	....	....
5841	32.2	5607	30.9	5625	31.0	5679	31.3	5265	29.0	4536	25.0	5787	31.9	5409	29.8
2457	13.6	1917	10.6	3337	18.4	3033	16.7	3150	17.4	3069	16.9	1944	10.7	....	....
5760	31.8	6138	33.8	....	....	....	....	....	....	....	....	....	....	....	....
4338	23.9	4482	24.8	....	....	....	....	....	....	....	....	....	....	....	....
−270	−1.5	−18	−0.1	....	....	....	....	....	....	....	....	....	....	....	....
18126	....	18126	....	....	....	....	....	....	....	....	....	....	....	....	....

Table XXXIII contains the results of the principal tests of the complete series, and in Fig. 177 are represented the fuel-consumption curves for four different speeds of the engine.

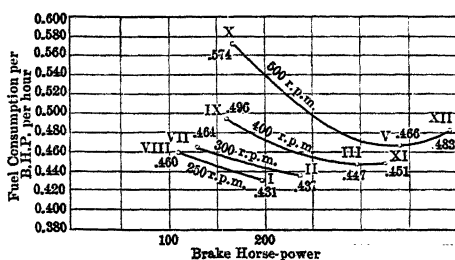



FIG. 177.—Fuel-Consumption Curves of Diesel 500 H.P. High-Speed Oil-Engine.

DUTY TEST OF A WESTINGHOUSE HORIZONTAL 500 HORSE-POWER DOUBLE-ACTING TANDEM ENGINE at the Plant of the Norton Co., Worcester, Mass.

Test made June 24 to 26, 1907, by G. J. Alden and J. R. Bibbins.

Cylinders: 23½ inches diameter,
33 inches stroke.

Bituminous gas-producer of the Loomis-Pettibone down-draft type.

A series of fuel-consumption tests on varying loads were made by observing the drop of the large gas-holder during the time each test was in progress. These tests are recorded in Table XXXIV, and graphically in Fig. 178.

There was also carried out a 51-hour test under average factory conditions, the general results of which are recorded in Table XXXV.

The mechanical efficiency of the engine was determined by comparing the results of 72 sets of indicator cards with the electrical output at the time the cards were taken; the generator efficiency having been carefully determined. The average mechanical efficiency was 83.5 per cent. Clearfield bituminous coal was used, the analysis of which averaged:

Volatile matter	19.87
Fixed carbon	73.71
Moisture	0.87
Ash	5.54
Sulphur	0.83
Heating-value, B.T.U. per pound, actual fuel	14.321
Heating-value, B.T.U. per pound, dry fuel	14.450

Per square foot of the producer fuel-bed there were gasified:

Maximum fuel	14.97 pounds per hour.
Minimum fuel	12.71 pounds per hour.
Average	13.33 pounds per hour.

The average gas-analysis during the 51-hour test was, approximately:

Carbon monoxide	23.5
Hydrogen	8.5
Methane	1
Nitrogen	63
Carbon dioxide	4
	100.0

Heating-value, by calorimeter.. 114.26 B.T.U. per cubic foot.
Heating-value, by analysis .... 112.40 B.T.U. per cubic foot.

The construction of the diagram, Fig. 178, is readily seen. Taking, for an illustration, the point A on the cubic feet gas per hour curve, which corresponds to a total gas-consumption of 30,000 cubic feet per hour, we obtain the corresponding heating-value consumed per hour, 3,190,000 B.T.U., as indicated on the B.T.U. per hour curve, and read on the B.T.U. per hour scale, near the right-hand side of the diagram. The horse-power corresponding to the point A, read on the horizontal bottom line of the diagram, is, approximately, 225 B.H.P. The heating-value consumed per brake horse-power per hour, therefore, 14,170 B.T.U., and the thermal efficiency approximately, $\frac{2{,}545}{14{,}170} = 17.9$ per cent. The latter two figures are read on, respectively, the B.T.U. per B.H.P. hour scale and the thermal efficiency scale, at the right-hand side of the diagram.

The divisions of the B.T.U. per B.H.P. hour scale and the thermal efficiency scale are such that the distance between each two successive horizontal lines represents, respectively, 800

TABLE XXXIV.

Fuel Consumption Tests on Various Loads—Holder Drop Test.

Number of Test.	A.	B.	C.	D.	E.	Remarks.
Duration of test—minutes	11	8	10	10	10	
Load rating per cent of full load	No load	25	45	70	Full load.	Circumference of holder 110.33 ft.
Speed of engine—revolutions per minute	158	156	154	152	150	
Brake horse-power	....	127.0	225.5	353.0	511.5	
Kilowatt	....	84.1	154.3	243.5	352.0	Average temperature of the gas 71.6° F.
Holder drop, ft. per hour	16.91	24.96	32.22	39.89	51.60	Barometer pressure 29.26 inches. Av. pressure of the gas $2\frac{7}{8}$ inches of water.
Gas consumption, reduced to 60° F. and 30 in. mercury:						
Total per hour, cubic feet	15760	23270	30050	37280	48200	Reduction factor 0.9642.
Per B.H.P. per hour, cubic feet	....	183.2	133.2	105.5	94.25	
Per kilowatt per hour, cubic feet	....	276.8	194.8	153.1	137.0	
Lower heating-value of the gas B.T.U. per cub.ft.	106.4	106.4	106.4	106.4	106.4	Average of all tests.
Heating-value consumed per B.H.P. per hour—B.T.U.	....	19480	14160	11215	10030	
Thermal efficiency referred to B.H.P.	....	13.05	17.96	22.68	25.36	

TABLE XXXV.

Average Heat Distribution During a 51-Hour Test (Av. 500 B.H.P.).

	Engine Only.	Entire Plant.
Useful work—B.H.P.	24.9	18.38
Friction and pump work	4.58	3.37
Loss in cooling water	34.22	25.22
Loss in exhaust and through radiation	36.3	28.81
Loss in producer		26.22
	100.00	100.00

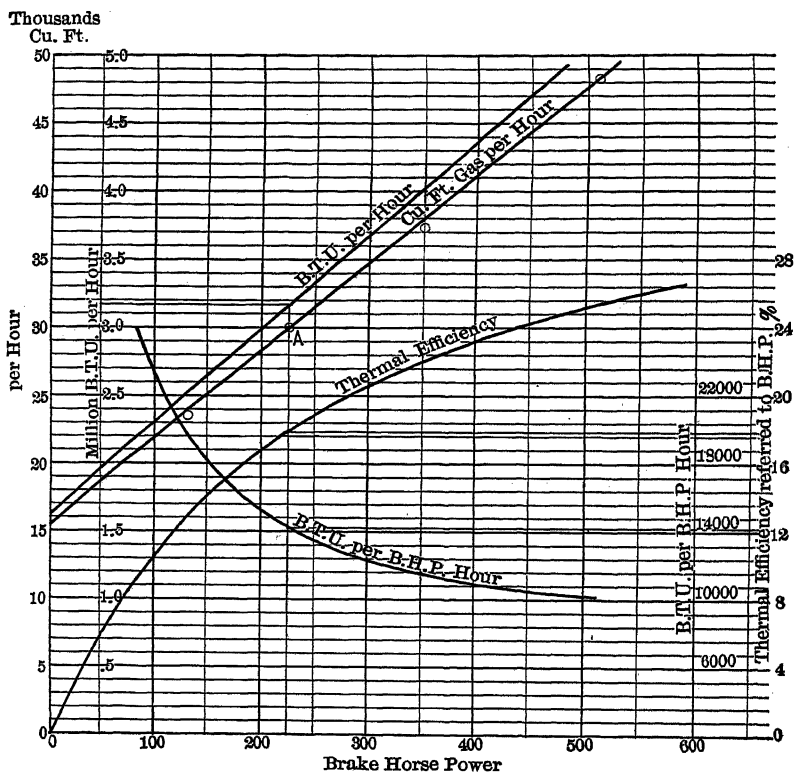



Fig. 178.—Efficiency Test of a Westinghouse 500 Horse-Power Gas-Engine.

B.T.U., and 0.8 per cent. The heating-value of the gas is figured at 106.4 B.T.U. per cubic foot.

TEST OF A NIEL SINGLE-CYLINDER 45 HORSE-POWER ENGINE on city gas. The test was made at Evreux, France, Nov. 11, 1901, by A. Witz and Moreaux.

The unit of work of the metric system is not exactly the same as that of the English; the English horse-power being approximately 1.4 per cent larger than the metric. This fact is often disregarded when data of the tests are transposed from one system to the other. To avoid discrepancies in the figures, however, due allowance should be made for the difference; or the fact that work is expressed in metric horse-power should be stated.

In the following test the data are given both in the original metric system and in the English, and, as a result of the discrepancy between the two horse-powers, it will be found that no two corresponding figures of the two columns referring to consumption per horse-power are, directly, reducible from one to the other.

Results of a Test of a Niel Four-Cycle Engine on City Gas.

	METRIC SYSTEM.	ENGLISH SYSTEM.
Diameter of the cylinder	350 m/m	13.78 in.
Length of the stroke	480 m/m	18.9 in.
A rope-brake was used, having:		
A sheave diameter	2m.220	84.7 in.
An effective diameter to the centre of the ropes	2m.254	88.74 in.
The circumference corresp. to the effective diam.	7m.081	278.75 in.
NO-LOAD TEST:		
Revolutions per minute	219.06	
Gas-consumption per hour	9.640 c.m.	340.4 c.f.
Barometric pressure	762 m/m	
The pressure of the gas (by water column)	36 m/m	1.42 in.
The temperature of the gas	10° C.	50° F.
Gas consumption per hour reduced to 32° F. and 760 m/m	9.331 c.m.	330 c.f.
Cooling water used per hour	255 litre	562 lbs.
Temperature of cooling water at inlet	12° C.	53.6° F.
Temperature of cooling water at outlet	67° C.	152.6° F.
Heating-value carried off by the water	14,025 cal.	55,640 B.T.U.

Results of a Test of a Niel Four-Cycle Engine on City Gas.—(*Continued.*)

	Metric System.	English System.
Full-Load Test:		
Duration of test	1 hour 45 m.	
Speed of engine (rev. per minute)	213.36	
Gas-consumption per hour	20.987 *c. m.*	741 *c. f.*
Barometric pressure	$763^m/_m$	....
The pressure of the gas (by water column)	$36^m/_m$	1.42 in.
The temperature of the gas	10.35° C.	50.63° F.
Gas-consumption per hour, reduced to 32° F. and $760^m/_m$	20.143 *c. m.*	711. *c. f.*
Cooling water used per hour	419 litre	924 lbs.
Temperature of the cooling water, at inlet	12.5° C.	54.5° F.
Temperature of the cooling water, at outlet	71.5° C.	160.7° F.
Heating-value carried off by the water per hour	24,721 *cal.*	98,129 B. T.U.
Net load on brake	137.083 *Kg.*	302.21 lbs.
Brake horse-power	46.02	45.38
Gas-consumption per B.H.P., gas of 32° F. and $760^m/_m$	0.439 *c. m.*	15.67 *c. f.*
Higher heating-value of the gas, at 32° F. and $760^m/_m$	5,664 *cal.* per *c. m.*	637 B.T.U. per *c. f.*
Heating-value consumed per hour per B.H.P.	2,486 *cal.*	9,981 B.T.U.
Heating-value corresponding to 1 horse-power hr.	632 *cal.*	2,545 B.T.U.
Heating-value carried off by the water, per B.H.P. per hour	537 *cal.*	2,162 B.T.U
Indicated horse-power	53.54	52.79
Mechanical efficiency	0.859	0.859
Gas-consumption per I.H.P. per hour	0.376 *c. m.*	13.47 *c. f.*

Distribution of the heating-value in one pound of gas:

Loss in the cooling water	0.216
Loss in the exhaust gases	0.388
Loss in friction and radiation	0.141
Converted into brake horse-power	0.255

The Prony Brake.—The brake referred to in the above engine-test consists of a brake-sheave around which is wound a rope of sufficient strength for absorbing the power of the engine. One end of the rope is fixed to a frame resting on the platform-scale

which measures the pull exerted in the rope, due to the between it and the brake-sheave, while the other end is t so as to produce the necessary frictional resistance. For ing of smaller engines the rope-brake is a suitable applia

In Fig. 179 is shown a design for a Prony brake, wl originally illustrated and described by Mr. Oliver (page the Transactions of the American Society of Mechani gineers, Vol. XXIV.), and which can be used successi absorbing much greater power than that for which the roj is suitable.

The brake-wheel is cooled by means of a stream c admitted between the flanges on the inside of the rim, effect circulation of the water, a scoop is arranged whicl the wheel revolves, scoops up and carries off the hot wate brake-shoes are built up of ribs of soft wood, poplar or ba with one-quarter of an inch space between the ribs, tc access for the lubricant underneath the brake-shoe. Th edge at the end of the brake-lever will rest on a platfor suitable for measuring the torque of the engine.

In order to simplify the figures involved in the calcul the power absorbed by the brake, the radial distance fr centre of the wheel to the knife-edge is preferably mac dimension such that the circumference described by this becomes an even number of feet.

Some Commonly Required Reduction-Factors.—For tions between the English and Metric Systems:

1 metre = 3.28083 foot = 39.37 inches.

1 foot = 0.304801 metre.

1 cubic metre = 35.315 cubic feet.

1 kilogramme = 2.2046 pounds.

1 pound = 0.4536 kilogramme.

1 kilogramme per square centimetre = 14.223 poun square inch.

1 pound per square inch = 0.0703 kilogramme per centimetre.

1 kilogramme-metre = 7.23292 foot-pound.

1 foot-pound = 0.1382 kilogramme-metre.

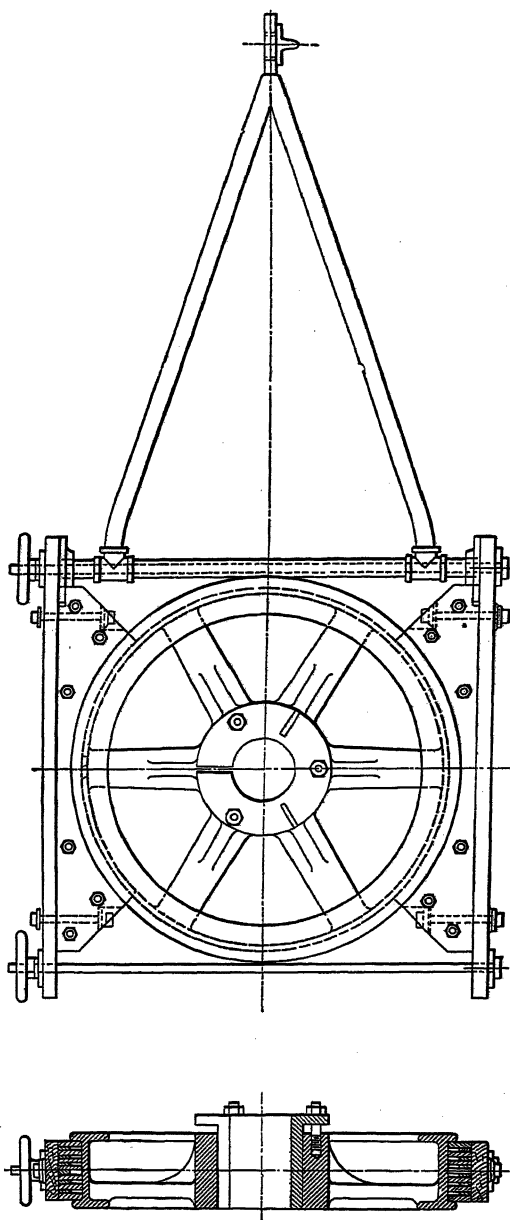



FIG. 179.—Prony Brake.

4500 kilogramme-metre per minute = 0.986 horse-power (English) = 1 horse-power (metric).

33000 foot-pound per minute = 1.014 horse-power (metric) = 1 horse-power (English).

1 calorie = 3.9683 B.T.U.

1 calorie per kilogramme = 1.8 B.T.U. per pound.

1 B.T.U. per pound = 0.555 calories per kilogramme.

1 calorie per cubic metre = 0.1124 B.T.U. per cubic foot.

1 B.T.U. per cubic foot = 8.91 calories per cubic metre.

Fuel consumption, in grammes per metric horse-power = 0.002235 pounds per English horse-power.

The mechanical equivalent of heat in the metric system is 427. Hence:

1 metric horse-power per minute = 10.54 calories per minute.

1 metric horse-power per hour = 632 calories per hour.

The Accelerating Force Due to the Reciprocating Parts.—The distance S from the cross-head to its head-end-centre position,

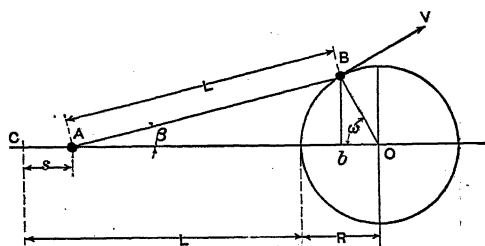



Fig. 180.

expressed in terms of the angular distance between the crank and the head-end dead centre is, according to Fig. 180

$$S = CA = CO - AO = (L + R) - (L \cos \beta + R \cos w)$$
$$= R(1 - \cos w) + L(1 - \cos \beta);$$

and we have $$\frac{R}{L} = \frac{\sin \beta}{\sin w},$$

thus $$\cos \beta = \sqrt{1 - \left(\frac{R}{L}\right)^2 \sin^2 w}.$$

If in the above square root we add the term $\left(\frac{R}{L}\right)^4 \sin^4 w,$

which is of inconsiderable magnitude compared with the other quantities of the root (its maximum value being $\frac{1}{256}$ when the maximum value of $\left(\frac{R}{L}\right)^2 sin^2\, w$ is $\frac{1}{16}$) then we obtain, approximately

$$cos\ \beta = \sqrt{1 - \left(\frac{R}{L}\right)^2 sin^2\, w + \left(\frac{R}{L}\right)^4 sin^4\, w}, \text{ or}$$

$$cos\ \beta = 1 - \tfrac{1}{2} \left(\frac{R}{L}\right)^2 sin^2\, w.$$

This value inserted in the expression for S gives

$$S = R\left(1 - cos\, w + \tfrac{1}{2}\frac{R}{L} sin^2 w\right).$$

The velocity of the cross-head is

$$v = \frac{d\, s}{d\, t} = \frac{d\left[R\left(1 - cos\, w + \tfrac{1}{2}\frac{R}{L} sin^2\, w\right)\right]}{d\, t}$$

$$= R\left(sin\, w + \tfrac{1}{2}\frac{R}{L} sin\, 2\, w\right)\frac{d\, w}{d\, t};$$

when t is a variable quantity representing time.

But as the velocity of the crank is assumed to be uniform, therefore

$$R\, d\, w = V\, d\, t,$$

$$\text{or } \frac{d\, w}{d\, t} = \frac{V}{R},$$

and hence

$$v = V\left(sin\, w + \tfrac{1}{2}\frac{R}{L} sin\, 2\, w\right).$$

The acceleration of the velocity is

$$a = \frac{d\, v}{d\, t} = \frac{d\left[V\left(sin\, w + \tfrac{1}{2}\frac{R}{L} sin\, 2\, w\right)\right]}{d\, t}$$

$$= V\left(cos\, w + \frac{R}{L} cos\, 2\, w\right)\frac{d\, w}{d\, t},$$

or by substituting $\frac{V}{R}$ for $\frac{d\, w}{d\, t}$ we get

$$a = \frac{V^2}{R}\left(cos\, w + \frac{R}{L} cos\, 2\, w\right).$$

The force required for giving this acceleration is

$$P = \frac{G}{g} a;$$

when G is the weight of the reciprocating parts and g the acceleration due to gravity.

Thus, for the forward stroke of the piston,

$$P_1 = \frac{G}{g} \frac{V^2}{R} \left(\cos w + \frac{R}{L} \cos 2w \right). \quad . \quad . \quad (100f)$$

For the return stroke we obtain

$$P_2 = \frac{G}{g} \frac{V^2}{R} \left(\cos w - \frac{R}{L} \cos 2w \right); \quad . \quad . \quad (100r)$$

if w be figured from the crank-end centre.

For the beginning of the stroke ($w = 0°$), $\cos w = 1$ and $\cos 2w = 1$, and for the end of the stroke ($w = 180°$), $\cos w = -1$ and $\cos 2w = 1$.

Hence we get, if r and l be expressed in inches:

The accelerating or retarding force at the head end of the piston stroke

$$P_1 = \pm \frac{12\,G}{g} \frac{V^2}{r} \left(1 + \frac{r}{l} \right), \quad . \quad . \quad . \quad (100h)$$

and the accelerating or retarding force at the crank end of the piston stroke

$$P_2 = + \frac{12\,G}{g} \frac{V^2}{r} \left(1 - \frac{r}{l} \right). \quad . \quad . \quad . \quad . \quad (100c)$$

At the point of the stroke where the connecting-rod forms with the crank a right angle, the crank-pin has momentarily a uniform and maximum velocity in the direction of the rod; the angle β is there a maximum, hence the velocity of the cross-head is, at the moment, uniform and a maximum. The velocity of the reciprocating parts, thus, changing at that point from an accelerating to a retarding one, the accelerating force is

$$P = \pm O.$$

On account of the approximation that has been introduced, this does not appear from the formulas $100f$ and $100r$.

The crank angle for $P = O$ is $tan\ a = \frac{v}{r}$.

The Tangential Crank-Effort.—Assume, according to the notations of Fig. 181, P to be the pressure acting on the piston when the crank stands in the position $O\ B$; Q the component of this pressure in the direction of the connecting-rod, and T the corresponding tangential crank-effort, then, it will readily be seen,

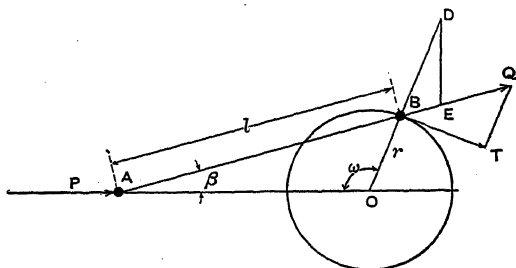



FIG. 181.

the relation between these three quantities, for any position of the crank, will be expressed by the equations

$$T = Q\ sin\ (w \pm \beta) \quad \frac{P\ sin\ (w \pm \beta)}{cos\ \beta} \qquad (127)$$

The sign of plus applying for all positions of the connecting-rod above the horizontal position and that of minus applying for its positions below the horizontal line.

The value of $cos\ \beta$ and β will be obtained from the ratio

$$\frac{r}{l} = \frac{sin\ \beta}{sin\ w}$$

Thus $$cos\ \beta \quad \sqrt{1 - \left(\frac{r}{l}\right)^2}$$

By inserting in equation 127 the values for P, w, β and $cos\ \beta$ the tangential effort for any crank-position may be solved.

When the tangential effort for a number of crank-positions is required, it is, however, much quicker work to derive the crank-effort by means of a graphical construction as follows:

Let it be required to find the tangential crank-effort for a crank-position $O\ B$; the pressure in the cylinder for the corre-

sponding position of the piston being P. The position of the cross-head pin, A, corresponding to the position, B, of the crank-pin we find by pointing off, from B to A, the length of the connecting-rod, l; and through the points A and B we draw the line $A\,B$ representing the centre-line of the connecting-rod, which, for positions of B to the right of the vertical centre line through O, we extend a proper amount beyond the point B.

On the crank-radius extended outside of the crank-pin circle point off, from B, the length $B\,D$ representing to a suitable scale the piston pressure, P, and draw from D, perpendicularly to the base-line $A\,O$, a line $D\,E$ terminating against the centre-line $A\,B$.

We have then

$$\frac{D\,E}{D\,B} = \frac{\sin\,(w\ +\beta)}{\sin\,(90 + \beta)},$$

or

$$D\,E = \frac{\sin\,(w\ +\beta)}{\cos\,\beta}\,P.$$

Hence

$$D\,E = T.$$

This construction may be carried out for any number of points of the crank-pin circle, and the required construction-lines for the various points will always be entirely clear of each other (Fig. 49).

Engine Belts.—Engine belts being, generally, as far as the designer of an engine has control, all of identical quality, and as they work, in most all cases, under practically the same conditions, the Nagle formula (see page 878, Kent's hand-book), can for convenience with respect to such belts be written

$$w = \frac{(7\ to\ 9)\ B.H.P.}{c\,V}; \quad . \quad . \quad . \quad . \quad (128)$$

w being the width of the belt, in inches,
V the rim-velocity of the belt-sheaves, in feet per second;
$B.H.P.$ the number of brake horse-power transmitted, and
c a coefficient.

If both belt-sheaves are of the same diameter then c becomes $= 1$; if not, c should be determined according to the number of degrees arc of contact the belt forms on the smaller sheave.

The value of c will vary in the following ratio:

Arc of contact between the belt and the smaller sheave,

90°	100°	110°	120°	130°	140°	150°	160°	170°	180°

coefficient c,

0.60	.65	.70	.75	.80	.85	.90	.94	.97	1.0.

For a normal belt speed, 80 feet per second, this formula allows, when the driving and driven sheaves are of equal size, 9 to 11 horse-power per each inch width of the belt; between which limits the actual allowance in successful belt transmissions is often found to lay. The choice between the wide and narrow belt must, of course, be made according to the importance of any particular case in hand.

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